IN TWO SECTIONS—SECTION TWO

TRANSACTIONS

of The American Society of Mechanical Engineers

SOCIETY RECORDS—Part III

Memorial Notices of Deceased Members

Part 1 of Society Records for the year 1937, containing Council and Committee Personnel and other general information, and the Membership List, was issued as Section Two of the Transactions for February, 1937. Part II, containing the Constitution, formed Section Two of the Transactions for July.

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Memorial Notices

THE purpose of Memorial Notices is to place on permanent record the biographical and professional data relating to deceased members of The American Society of Mechanical Engineers. Hence every effort is expended to insure accuracy, and to make the notices as inclusive as is reasonably possible.

The first source of information upon which these notices are based is the Society's file of membership applications and transfers. In the case of the more recent member, these application records are fairly complete. The applications of those who became members many years ago, however, contain less detailed data, and in many cases the sponsors are no longer alive, so that it is difficult to obtain assistance from this source. If the member has been retired for several years prior to his death, his business associates are frequently hard to locate, and, in some cases, members of his family cannot be found. While all these factors add to the difficulty of obtaining accurate and fairly complete data, every possible source of information is explored, with the result that publication of the notice is sometimes delayed.

It is the practice of the Committee on Publications in the case of many deceased members to ask former friends and associates to prepare the obituary. The object is to secure a final record that will be more valuable for having been prepared by men who knew the deceased and are competent to evaluate his work. Memorials prepared in accordance with this recent policy are signed by those who wrote them or who collaborated in their preparation. To all persons who have thus cooperated, the Committee acknowledges its gratitude.

The Committee has also established most helpful and cordial contacts with other societies in the preparation of these notices. As members of this Society are sometimes members of others also, collaboration in the preparation of obituaries holds the possibility of securing a more completely authoritative record. Several obituaries published with this series will be found to have been republished in full from the records of other societies, or to have been adapted from these records. To these societies, and to their members who prepared the obituaries so used, the Committee wishes to express its thanks.

The Committee also appreciates and acknowledges the assistance that has been given by relatives, business associates, and friends in the preparation of all other memorials. It also acknowledges its debt to such sources as Who's Who in Engineering, Who's Who in America, and similar publications; the Encylopedia of American Biography, National Cyclopedia of American Biography, and New International Year Book; the technical and daily press; colleges and universities and their alumni associations; and to engineering and other societies which have supplied information from their records.

Relatives, business associates, and Local Section and Student Branch officers are urged to notify the Society promptly of the deaths of members. Newspaper clippings or obituaries in any other form should be sent whenever available and the names and addresses of those who can supply further information should be furnished. A special form for supplying complete details will be forwarded by the Office of the Society upon request.

THE COMMITTEE ON PUBLICATIONS

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Memorial Notices

CHARLES GEORGE ADSIT (1874–1935)

Charles George Adsit, president of the Des Moines Railway Company and of the Des Moines & Central Iowa R.R., died on March 27, 1935. He is survived by his widow, Helen (Brinkman) Adsit, whom he married in 1907, and by his son, Charles G. Adsit, Jr.

Mr. Adsit was born at Ironton, Ohio, on November 1, 1874, son of Byron De Witt and Emma (George) Adsit. After completing his high school course in 1892 he entered the office of John R. Allen, mechanical engineer, of the firm of Ball & Allen, consulting engineers, Chicago, and studied under his direction; in 1893-1894 he also took a course in mathematics at the University of Chicago. His association with Mr. Allen terminated in 1894 when the latter took a professorship at the University of Michigan. During the next two years he engaged in engineering and contracting business in Chicago and in 1897–1898 he was with a mining party in Alaska. Upon his return to Chicago he became connected with the firm of Rockwell & Snyder, contracting engineers, in charge of municipal engineering work, principally at Waupun, Wis.

In 1901 Mr. Adsit entered the electrical field in the testing department of the General Electric Company at Schenectady, N.Y. After about two years' experience there he went to Breckenridge, Colo., to take charge of the design and construction of a hydroelectric plant for the Gold Pan Engineering & Mine Supply Co. In 1904 he was chief engineer in charge of the design and construction of a hydroelectric plant for the Kimberly Montana Coal Mining Company, at Jardine, Mont. He also changed the drive of their mining and reduction works from steam to electrical operation.

For five years, beginning in 1905, Mr. Adsit was consulting electrical and mechanical engineer for the Black Mountain Mining Company, of Chicago and Magdalena, Sonora, Mexico. He designed and constructed a steam turbine generating station transmission system from the railroad to the mine, some forty miles distant, and applied electric drive to all of the mining and reduction equipment. This was one of the largest mining operations in Mexico at the time.

Mr. Adsit turned to railway engineering in 1908 when he became consulting engineer for the Warren Company and the Warren-Bisbee Railway, at Bisbee, Ariz. He designed and constructed an interurban electric railway system, operating in the Bisbee mining district under conditions made difficult by the mountainous character of that section of the country. Upon the completion of this work in 1911 he took a position as engineer for the Northern Contracting Company, and as such was in charge of the hydroelectric development at Tullulah Falls, Ga., for the Georgia Railway & Power Co. He remained to serve as chief engineer, in charge of further development work of the company, and from 1923 to 1928 was vice-president and executive engineer of the company, and its successor, the Georgia Power Company. He resigned to engage in consulting practice in Atlanta, Ga., until he became president of the Des Moines company and railroad in 1929.

Mr. Adsit belonged to three of the national engineering societies, the A.S.C.E., A.I.E.E., and A.S.M.E., becoming a member of the A.S.M.E. in 1927. He was a Fellow of the A.I.E.E., served as a vice-president of the Institute in 1921–1923, and was active in its committee work. He was also a member of The Franklin Institute.

ST. GEORGE MASON ANDERSON (1873–1936)

St. George Mason Anderson, superintendent of the Tredegar Company's foundry and machine shops, died in his native city of Richmond, Va., on February 24, 1936. He was born on June 27, 1873, son of Archer and Mary Anne (Mason) Anderson. After preparing for college at McGuire's School in Richmond and the University School in Petersburg, Va., he entered Stevens Institute of Technology, Hoboken, N.J., in 1890 and was graduated with an M.E. degree in 1894.

Upon graduation, Mr. Anderson was appointed assistant chemist in the Virginia State Department of Agriculture and held that position for approximately one year. In November, 1895, he entered the employ of the Tredegar Company, Richmond, Va., as assistant superintendent of the horseshoe department and remained with that organization until the time of his death 41 years later. Mr. Anderson was promoted to the superintendency of the spike and bar mills about 1899 and, approximately eight years later, was made superintendent of the company's rolling mills, foundries, and machine shops. In this capacity, he had supervision of more than fifteen hundred men engaged in the manufacture of a wide variety of products which included railroad and boat spikes; horse and mule shoes; angles; fishplates; bars; car wheels; railroad, agricultural, architectural, and general castings; target shells and proof shot for the U.S. Army and Navy; steel shells for the U.S. Navy; shrapnel cases for the U.S. Navy, British Admiralty, British War Office, and Russia; and blooms for its own use. Since January, 1923, when the rolling mills and the horseshoe department were placed in charge of a separate superintendent, Mr. Anderson had been in charge of the foundry and machine shops.

He joined the A.S.M.E. as a member in 1919. He is survived by three daughters, Penelope (Anderson) McBride and Mary Mason and Margaret Ward Anderson, and by two brothers, Archer, Jr., and J. R. J. Anderson, who are president and treasurer, respectively, of the Tredegar Company. His wife, Penelope (Weddell) Anderson whom he married in 1898, died in 1925.

SIR JOHN ASPINALL (1851–1937)

Sir John Audley Frederick Aspinall, engineer, executive, educator, author of many outstanding papers relating to technical achievements of the British Railways with which he was associated, man of grace and charm, beloved by all who knew him, was born at Liverpool on August 25, 1851, the son of John Bridge Aspinall, Queen's Counsel. His death occurred on January 19, 1937, bringing to a close a life of notable contributions in railway engineering progress.

Throughout his busy career, Sir John was identified with the development of the reciprocating steam locomotive and the shops and associated facilities for its maintenance. He advocated the tonnage train with locomotives of great power. A technical man, he possessed such breadth of vision that his interests and capabilities extended far beyond the mechanical field. In recognition of his competence in matters pertaining to traffic and operating practice, he was appointed general manager of the Lancashire and Yorkshire Railways in 1899, the first man of technical training in British railway history to serve beyond the mechanical departments. Having already established the economy of the heavy locomotive, he now advocated and installed in 1904 the first main line electrification in the United Kingdom with the conversion of the congested line, Liverpool to Southport.

As a youth of extraordinary native ability, after acquiring the general education afforded by Beaumont College, Berkshire, he became indentured apprentice to John Ramsbottom and was later, in 1871, transferred to serve F. W. Webb upon the retirement of Ramsbottom as locomotive superintendent of the London and North Western Railway. Completing his term of apprenticeship at the age of twenty-one, Aspinall was appointed assistant manager of the Crewe steel works, subsequently being elevated rapidly and successively to the positions of, first, manager, then locomotive engineer, and later locomotive superintendent at Inchicore, well-known works of the Great Southern and Western Railway of Ireland; chief mechanical engineer, and later general manager, of the Lancashire and Yorkshire Railway; consulting mechanical engineer to the Ministry of Transport; and consultant to the newly organized London, Midland and Scottish Railway, 1924 to 1926.

As chief mechanical engineer of the Lancashire and Yorkshire, Aspinall was given opportunity to put into practice many of the ideas and advancements in railway engineering technique which he considered of practical value but which were judged to be revolutionary at the time. He constructed the works at Horwich, instituting a planned, progressive system of locomotive repair which was to be adopted quite generally throughout the Empire and copied by many railways in other countries. Notable among his achievements as chief mechanical engineer are his proof of the practicability of a high center of gravity in steam locomotives, and his unremitting efforts toward standardization and interchangeability of parts between locomotives of various classes, the latter being a project of great worth at the time, 1886 to 1899, a period during which highly specialized locomotive types were not required by traffic specifications. He investigated locomotive resistance and pioneered in the study of air flow and streamlining, recognizing the influence of the atmosphere in high-speed running.

As general manager, in addition to his original efforts in encouraging electrification to increase track capacity, he effectively advanced the cause of consolidation of competitive railway systems to the ultimate advantage of all. No end of credit must be accorded him for the courage and foresight which he displayed in abandoning precedent against long odds and established opinion to introduce daring new practices which, in his judgment, offered promise of new advantages, later to be generally recognized and applauded.

Perhaps mindful of the benefits of competent early training as they

served him so well, Sir John maintained a deep sympathy for the young men who entered the railway field under his direction. He solicited funds and personally outlined the plans for the establishment of the Mechanics Institute at Horwich, a school of practical training which has grown steadily in size, facilities, compass of its activities, and prestige. His interests in educational work secured for him the chair of associate professor of railway engineering at Liverpool University, in 1902, to which office he contributed generously, principally in an advisory capacity.

Sir John Aspinall was variously honored. Knighthood was conferred upon him by the King in 1917 in recognition of his many contributions to transport progress throughout the British Empire—for his influence was far-reaching. He was awarded Watt medals by both the Institution of Civil Engineers and the Institution of Mechanical Engineers, and the Telford premium, by the Institution of Civil Engineers. He served most graciously as host to our own Society when its representatives met jointly with the Institution of Mechanical Engineers in London in 1910, at which time he was president of the Institution. In recognition both of his achievements in the technical field and his efforts toward consolidating the endeavors of the two national mechanical engineering societies, he was awarded honorary membership in the A.S.M.E. in 1911. He had also been honored with the presidency of the Institution of Civil Engineers and the Institution of Civil Engineers of Ireland.

Sir John Aspinall devoted sixty crowded years to the solution of rail transport problems. He was progressive beyond the age in which he lived. The principles which he instituted many years ago and for which he labored with untiring zeal have since been recognized by virtue of their accomplishment and, through them, Sir John will live for many years as a guiding influence in railway circles the world around.—[Memorial prepared by L. K. Sillcox, Watertown, N.Y. Mem. A.S.M.E.]

GEORGE MILLER BARTLETT (1873–1936)

George Miller Bartlett died on September 16, 1936, at the Methodist Hospital, Indianapolis, Ind., after an illness of less than three weeks. He was born on August 5, 1873, in Chicago, Ill., son of Abner Dearborn and Sarah A. (Miller) Bartlett. He was married in 1905 to Mary L. Doughty, of Brooklyn, N.Y., who, together with one son, Dr. Paul D. Bartlett, of Harvard University, survives him.

Professor Bartlett's preparation for his profession was unusually sound. His college work at Amherst College was preceded by several years of practical experience so that his Bachelor of Science degree (1901) was received at the age of twenty-eight, with major subjects, physics and mathematics. Four years later (1905) he received the Master of Arts degree for work in mathematics.

After graduation he taught for nine years; two at the Case School of Applied Science followed by seven at the University of Michigan. During this period he produced two textbooks: one on Descriptive Geometry and one on Graphical Statics of Machines.

In 1910 he became chief engineer of the Diamond Chain & Manufacturing Co. of Indianapolis, and continued to serve that company as consulting engineer after being called to Purdue University as associate professor of machine design in February, 1927. He was advanced to professor in 1931.

Professor Bartlett was an inventor and writer quite above the ordinary and some of his papers are landmarks in engineering and educational practice. He confidently believed that invention could be taught and in 1934 instituted a forum on the technique of invention, followed in 1935 by regular class instruction on the subject. His inventions included: Basic helicoidal rack for the generation of teeth for hyperboloidal gears; two forms of universal joints for constant ratios; two types of flexible shaft couplings; universal angular transmission; chain-drive planetary reduction unit; automatic oiling device for chain cases; reciprocating motion from "wabble ring;" "lineometer" for determining chain and belt lengths; and numerous attachments to special machinery used in the manufacture of roller transmission chains.

Unusual richness distinguished Professor Bartlett's social life. As guest, host, comrade, and friend he was always tactful, gracious, genial, and true, and his fine background of literary, musical, and dramatic knowledge gave his conversation unusual interest and character. The varied nature of his interest is indicated by his affiliations with organized groups. He had been a member of the A.S.M.E. since 1911. He was also a member of the Society for the Promotion of Engineering Education, American Gear Manufacturers' Association, Indianapolis Literary Society, Scientech Club of Indianapolis, Delta Upsilon fraternity, and Pi Tau Sigma, honorary mechanical engineering fraternity. His long affiliation with the Congregational Church in Indianapolis was transferred to the Trinity Methodist Episcopal Church when he went to Lafayette.

Professor Bartlett had been one of the official representatives of the A.S.M.E. on the Sectional Committee on the Standardization of Transmission Chains and Sprockets since its organization in 1917, and had served as secretary of the sectional committee and as chairman of its Subcommittee on Roller Chain Standardization since that date. For two years prior to his death he also represented the American Gear Manufacturers' Association on the sectional committee, and he was also chairman of the A.G.M.A. Sprocket Committee at the time of his death.

As a teacher he was painstaking and thorough, demanding from the student something of the same earnestness of purpose which so characterized himself, an attitude which never fails its purpose.—[Memorial prepared by Prof. L.V. Ludy, Purdue University. Mem. A.S.M.E.]

GEORGE BARTOL (1857–1936)

George Bartol, whose parents were George Murillo and Elizabeth (Washburn) Bartol, was born on May 16, 1857, at Lancaster, Mass., where he secured his early eduction. He entered the Massachusetts Institute of Technology with the Class of 1877, specializing in mining engineering, and after securing his B.S. degree, took postgraduate work in chemistry for a year. He entered the employ of the Otis Iron & Steel Co., Cleveland, Ohio, in February, 1879, as a chemist. Following his death on April 3, 1936, the following tribute was published by The Otis Steel Company.

"Few men have the privilege of participating in the growth and development of any institution over a period of 57 years. That was the opportunity which George Bartol earned in the years which began in 1879 when, fresh from Massachusetts Institute of Technology, he came to the Otis Steel Co.'s pioneer open-hearth furnace. His first job was as a chemist, to analyze molten metal which was setting the pace for a nation's industry. During the years which followed he was one of the key men in all production operations. Later as vice-president and president of the company, he endeared himself to every man in the organization.

"Notwithstanding his inactive part in affairs of the company in the last 11 years, Mr. Bartol's constant interest and regular visits which always carried him into every part of the mill, kept him close to all of us. The word of his death on Friday, April 3, after a severe attack of influenza, meant one thing to all—that a lovable, democratic gentleman, whose character and friendliness were known across the seas as well as through the Cuyahoga Valley, had made his last friendly trip through the mills.

"When ownership of the Otis properties shifted to England back in 1889, after the founders' fame as producers of basic open-hearth steel had spread abroad, it was George Bartol who won the confidence of the new owners. In 1897 they named him the American manager, and a year later elevated him to the presidency of the company. For nearly thirty years he continued to direct the company's destinies. Expansion during this period was rapid. By 1912 Otis had outgrown the Lakeside site and Mr. Bartol and his colleagues bought the valley site which today is the scene of our Riverside operations. Having directed the expansion during the years up to 1925, and looked to for counsel even in the years which followed, Mr. Bartol knew intimately every nook and corner of the far-flung Otis works. In recent years his habit never varied from a routine which brought him into the office building to get his mail and then sent him on a tramp clear through the mill where he saw everything, talked to everyone, and always came away with a faithful picture of what was going on.

"He was an old-fashioned boss with whom few could compare even before big mills came into being. Known to us all and to two generations who preceded us, Mr. Bartol will be remembered and missed by all of the Otis family, which is joined with his very own by sympathy."

He is survived by two daughters, Mrs. I. F. Weidlein and Miss Elizabeth Bartol, both of Cleveland. Mrs. Bartol died in 1927.

Mr. Bartol had been a member of the A.S.M.E. since 1886, and he had belonged to the Cleveland Engineering Society since 1880, being one of its early members.

JAMES VERNON ROWLAND BIBBINS (1877–1936)

James Vernon Rowland Bibbins (more generally known as J. Rowland Bibbins) died on January 27, 1936, in Washington, D.C., where he was traffic engineer in charge of local traffic signal and transit surveys.

Mr. Bibbins was born on May 25, 1877, at Williamsport, Pa. He attended the Baltimore (Md.) City College and was graduated from the University of Michigan in 1899 with a B.S. degree in electrical engineering. During his college years he had spent a summer in general engineering work for the Fuller Electric Company, Detroit, Mich., and for several years after his graduation he worked in that

city for the Detroit Edison Company, the Public Lighting System, and the Detroit United Railway. He then went to East Pittsburgh, Pa., to do technical writing for the Westinghouse Machine Company. He continued in the service of the Westinghouse companies for some years, preparing various analyses and reports, and, in the position of commercial engineer, handling matters involving economic research in the field of steam and gas power development.

In 1909 Mr. Bibbins became associated with Bion J. Arnold, consulting engineer of Chicago. For more than ten years he worked with Mr. Arnold on transportation and related problems of many large cities in the United States and Canada, including transit signal and traffic plans, port and terminal developments, and economic policies involved therein, particularly valuation.

Either as Mr. Arnold's principal assistant or later on his own behalf, Mr. Bibbins was in charge of numerous technical investigations and reports to state, municipal, and other public bodies and to various corporations. He conducted subway cost surveys for Chicago and Philadelphia, and a transportation survey for the Port of New Orleans, contributed data in connection with terminal arbitration in Chicago, Syracuse, and Baltimore, made valuations of properties in New York and Brooklyn, and developed transit plans for San Francisco, Pittsburgh, Detroit, Montreal, Winnipeg, Denver, and many other cities.

He resigned as supervising engineer of the Arnold Company in 1921 to become manager of the new Department of Transportation and Communication of the United States Chamber of Commerce, at Washington. This Department was created to cover shipping, ocean and inland, steam and electric railroad transportation, air transportation, cables and telegraphs, postal facilities, and highways. It was organized under the supervision of Mr. Bibbins and directed by him for about two years. After that he went into business for himself as a consulting engineer, with headquarters in Washington.

In 1925 he was engaged by the city of Philadelphia as a consultant in a rate case between the Public Service Commission of Pennsylvania and the Philadelphia Rapid Transit Company. Later he was also retained by the city of Philadelphia to formulate a contract for the rental of the city-owned subway to the Philadelphia Rapid Transit Company. In 1928 he was engaged by the Montreal Tramways Company to report on the operation of the system and to make recommendations for its future development. In 1929 he was retained by the Third Avenue Railway Company in New York to make studies of the effect of traffic lights upon the operation of the street railways and to make recommendations for the timing of the lights so as to get greater street capacity, both for vehicles and street cars. He made a 25-year economic forecast of central building development for Detroit, a railway merger routing plan for Washington, and traffic control thoroughfare plans for Indianapolis, Lancaster, and other cities.

Mr. Bibbins became a junior member of the A.S.M.E. in 1904 and a member five years later. He served as chairman of the Gas Power Section of the Society in 1910, and was a member of a special committee on the standardization of catalogs from 1911 to 1914. He was also a member of the American Institute of Electrical Engineers and past-chairman of its Chicago Section. He had belonged to the Western Society of Engineers, Society of Terminal Engineers, American Electric Railway Association, and American Association of Port Authorities, and had served on several terminal committees. He was deeply interested in nation-wide industrial research and, when serving as president of the Michigan Engineers, helped to formulate a plan for the establishment of a Department of Engineering Research at the University of Michigan.

At the time of his resignation from the Arnold organization, a statement concerning his achievements which was issued included the following paragraph:

"Author of many technical papers and addresses on railway development, valuation, and power production, Mr. Bibbins has always sought to dignify his profession by securing fair, adequate discussion and thorough understanding of its problems. His recent contribution, 'City Building and Transportation,' has carried this spirit into the comparatively new civic field. He is a firm believer in the usefulness of engineers in public affairs, and the broad forum offered by the U.S. Chamber of Commerce will make this possible to an unusual degree. It is his hope that, ultimately, the technical talent of the country may be mobilized and effectively directed toward this great end of united public service."

Mr. Bibbins was an accomplished singer and the author of several volumes of verse published under the name of James Robbins.

LEE JACKSON BLACK (1870–1934)

Lee Jackson Black, manufacturer and inventor, was born at Jefferson, Tex., on June 24, 1870, son of Oliver Hazard Perry and Mary Kearny (Hunt) Black. Mr. Black received his early education in various country schools in Texas and was graduated at Austin College, Sherman, Tex., in 1893. Subsequently he spent a year as bookkeeper for a lumber company in Fannin County, Tex., and then was foreman of a dredging crew at Sabine Pass, Tex. For two years thereafter he was associated with a creosoting company at Beaumont, Tex., and from 1896 to 1901 was employed by the Hanson Construction Company, a bridge building concern, at Jasper, Tex., for which he supervised the construction of bridges throughout the Southwest and had charge of the building of a railway from Beaumont to San Augustine, Tex. He entered the oil industry in 1901 as general superintendent of the Higgins Oil & Fuel Co., at Beaumont, and continued in that position until 1911, when he resigned to assume active charge of the Beaumont Iron Works which, with others, he had purchased from a receiver in the preceding year. Impressed with the need for a plant specializing in the production of oil field equipment, he limited the output of the Beaumont Iron Works to the requirements of the rapidly expanding oil industry, and the business was a success from the outset. Under his management as vice-president and general manager, 1912–1914, as president, 1914–1926, and as chairman of the board, 1926 until his death, the Beaumont Iron Works Company developed into a $700,000 corporation and one of the leading concerns of its kind, its products being extensively used in oil fields throughout the world. To this achievement he contributed not only by his organizing and administrative ability but with his engineering and inventive genius as well. During his professional career he received more than one hundred patents, of which he assigned forty-eight to the Beaumont Iron Works during the last ten years of his life. These related chiefly to oil-well-drilling machinery, but inventions applicable to various other industrial activities were conceived and manufactured by him. Black was the first to make a practical adaptation of the principle of cooling brake drums with a flow of water. This he incorporated in the so-called "Dreadnaught" water-cooled draw works, the most widely known product of his creative skill. Drilling rigs embodying this design were sold in all parts of the world. Among his other more important inventions were the straight-line crown block, in which all the line sheaves are placed in a single shaft, an improvement that won wide attention in the engineering world; his "grease pin" shaft which permits the entire working assembly of a crown block or traveling block to be lubricated simultaneously by a quarter-turn of a lug; and his "straight shot" sludge pump which became standard in the oil fields. He also invented an all-steel internally sway-braced walking beam, which came into extensive use; a long-stroke pumping jack for deep-well use in conjunction with a central power plant, and a "shotgun feeder," a widely known sawmill device whose safety and efficiency won for it universal acceptance in the lumber industry. Shortly before his death he obtained a patent on an air-conditioning device for use in private houses.

He became a member of the A.S.M.E. in 1923 and belonged to the Masonic order and the Beaumont and Rotary clubs of Beaumont. Interested in civic affairs, he served for several years as water-works commissioner and in that capacity designed a special filtering plant and redesigned and improved Beaumont's water system. He possessed an energetic, dominating personality and a genial, kindly nature, and was especially interested in the education of young people, a number of whom were enabled to attend institutions of higher learning through his generosity. He was married on January 30, 1901, to Maude, daughter of John Randolph Bevil, of Kountze, Hardin County, Tex., and died, without issue, at Beaumont, Tex., on August 10, 1934.—[From the National Cyclopedia of American Biography.]

WESTINUS BOER (1894–1937)

Westinus Boer, who died suddenly at his home in Groton, Conn., on February 16, 1937, was born on April 27, 1894, at Delfs Haven, Rotterdam, Holland. His technical education was secured through the Rotterdam Academy of Plastic Arts and Technical Science. During the years 1916 to 1922, he was employed by the Verschure and Company's Shipyard and Engine Works, Amsterdam, Holland, in the design of marine steam engines and boilers.

In the year 1923, he came to the United States and obtained employment with the Worthington Pump & Machinery Co. at Cambridge, Mass. During the early part of 1924 he was connected, for a short time, with the Bethlehem Shipbuilding Corporation, at Quincy, Mass.

In April of the same year, he accepted a position with the Electric Boat Company, Groton, Conn., with whom he remained until his death. During this period, he was employed as draftsman, designer, and later was in charge of the Design and Research Calculating Department of the Mechanical Division of this company. Mr. Boer became a citizen of the United States in 1929 and a member of A.S.M.E. in 1932. He served on the Executive Committee of the Norwich Local Section in 1933-1934.

On March 22, 1930, he obtained a U.S. patent for "a propelling device for air and water craft." The purpose of the device was to provide a vertical lift for airplanes as well as forward thrust, resulting in less resistance to the craft in motion. The device was also patented in Canada and several other countries.

Mr. Boer is survived by his widow, Dorothy (Foley) Boer, whom he married in 1928, and by a son and a daughter, Westinus, Jr., and Gertrude.—[Memorial by William Ward, Chief draftsman, Electric Boat Company, Groton, Conn.]

FRANK R. BREHM (1882–1935)

Frank R. Brehm, who became a member of the A.S.M.E. in 1927, was born on December 4, 1882, at Franklin, Ohio. Supplementing his early public and high-school education he took courses through the International Correspondence Schools, securing a diploma in mechanical engineering.

At the age of twenty-five Mr. Brehm became master mechanic for the Dayton Rubber Manufacturing Company, Dayton, Ohio. He left there in 1912 to serve as superintendent of the boiler plant of the National Cash Register Company, Dayton, where he had charge of about fifty men. The plant consisted of 14 1000-hp boilers, vacuum and water pumps, water-softening apparatus, ash-handling machinery, and other equipment, and was regarded as one of the most up-to-date in that section. Mr. Brehm continued in that work for three years, then went to Hamilton, Ohio, as erecting engineer for the Hooven, Owens & Rentschler Co. During his three years with this company he installed engines of from 500 to 5000 hp, of the cross-compound, tandem-compound, and triple-expansion types.

From 1918 to 1924, Mr. Brehm was with the American Rolling Mill, at Middletown, Ohio. Here he was assistant to the maintenance engineer, having charge of repairs in the boiler house and engine room, and of locomotive cranes. During the following year he was chief engineer for the Middletown Artificial Ice Company, for whom he planned and built a new engine room and boiler house.

In 1925 he became superintendent of erection for the Conveyors Corporation of America of Chicago, Ill., which subsequently was absorbed by the United Conveyor Corporation. Mr. Brehm directed the installation of various types of ash- and coal-handling machinery, and also made designs for this kind of equipment.

After leaving this company he was employed as superintendent of the boiler room of the Marathon Paper Mills Company at Rothschild, Wis., from early March to the end of September, 1928. Subsequently he was chief engineer for the Combined Locks Paper Company, with which, so far as can be determined, he was associated at the time of his death on May 15, 1935. He is survived by his widow.

MAURICE LEWIS BURGHAM (1901–1935)

Maurice Lewis Burgham, sales engineer with the Edgewater Steel Company of Pittsburgh, Pa., died at the Homeopathic Hospital in that city on July 19, 1935, from pneumonia. His widow, Esther (Bauman) Burgham, survives him.

Mr. Burgham was born at Parnassus, Pa., on April 3, 1901, son of Edwin S. and Estella (Bevan) Burgham. He was graduated from the University of Pittsburgh with a B.S. degree in mechanical engineering in 1923. For nine months after graduation, he was an assistant to the test engineer at the Colfax station of the Duquesne Light Company and made daily calculations of boiler efficiencies, water rates, heat transfer, and similar data. A brief engagement with the Blaw-Knox Company, Blawnox, Pa., as mechanical draftsman, detailing and designing equipment for blast and open-hearth furnaces followed, and, from 1924 to 1926, he was a draftsman and mechanical engineer with the Aluminum Seal Company, New Kensington, Pa., where he did detailing and designed machines. From 1926 until his death, Mr. Burgham was employed by the Edgewater Steel Company, first as a draftsman and designing engineer and, since 1930, as a sales engineer. Much of his work was in connection with design, development, and promotion of sales.

Mr. Burgham became a junior member of the A.S.M.E. in 1925. He was also a member of the Railway Club, Pittsburgh Athletic Association, and Pitt Engineering Association, and held a commission as second lieutenant in the Ordnance Reserve Corps, U.S. Army.

EDWARD PARKER BURRELL (1871–1937)

Several of the world's greatest telescopes, together with modern turret lathes in active service in industry, are lasting monuments to Edward Parker Burrell, director of engineering for The Warner & Swasey Co. from 1924 until his death on March 21, 1937, in his sixty-seventh year.

Associated with The Warner & Swasey Co. since 1900, he was credited as the developer of hexagonal turrets, which greatly increase the general adaptability and uses of machine tools. One of the pioneers in building gear trains for lathe heads that run entirely in oil, he added greatly to the life and efficient operation of machine tools. These contributions to modern production equipment, together with telescopes designed by Mr. Burrell, the last of which is at the McDonald Observatory in Texas, testify to Mr. Burrell's lasting influence upon science and industry.

Mr. Burrell was born in Hall, N.Y., on February 11, 1871, the son of Edward and Elizabeth (Parker) Burrell, and was graduated from Cornell University as Mechanical Engineer in Electrical Engineering in 1898. He received a Master's degree there the following year and immediately thereafter joined The Warner & Swasey Co. He was married in 1904 to Miss Katherine Ward, of Cleveland, who survives him.

At Warner & Swasey, his first task was in the development of instruments for sending and receiving cable and telegraph dispatches by a new automatic method. Electrically motivated brushes which punched holes (like those in a player piano roll) in tape, at high speed, achieved the desired speed-up in transmission of messages, foretelling recent application of comparable systems for teletype and other telegraphic transmissions.

More than thirty years devoted to the design of telescopes and turret lathes followed this initial task. Early in his career as a telescope builder, Mr. Burrell developed a range finder which compensates for the bending of light rays in the atmosphere. A relocating device for Coast Artillery range finders, also developed by him, has aided naval defense. A universal mounting adaptable to each of seven types of telescopes in use aboard American battleships at the outset of the World War was designed by him, as well as a back sight for artillery.

More than twenty important telescopes and domes built by The Warner & Swasey Co. reflected the workmanship and ideas of Mr. Burrell. His was the design for the 72-in. reflecting telescope for the Dominion Astrophysical Observatory at Victoria, B.C., the 69-in. reflecting telescope for Ohio Wesleyan University at Delaware, Ohio, and the new McDonald 82-in. reflecting telescope in western Texas. The latter, with two "pulpits" attached inside to either end of the observing bridge, will afford astronomers direct access to any point within the dome, for purposes of observation. With the "pulpit" floors also adjustable, astronomers will be able to reach special observing positions as readily as they can change auxiliary mirrors and equipment.

Mr. Burrell became a member of the A.S.M.E. in 1919. He had served as a member of the Technical Committee on Standardization of T-Slots since its organization in September, 1924, and on the Technical Committee on Standardization of Tool Posts and Shanks from its organization in January, 1926, to December 3, 1935, in both cases as a representative of the National Machine Tool Builders Association. He also belonged to the Cleveland Engineering Society and Sigma Xi. In June of 1936 the degree of Doctor of Engineering was conferred upon him by Case School of Applied Science.

Mr. Burrell's death removes an important figure; one who found his opportunities in labor for world science and industry through a company whose 90-year old founder, the late Ambrose Swasey, was a life-long inspiration.

ARTHUR DUDLEY BUZBY (1889–1935)

Arthur Dudley Buzby, a member of the four national engineering societies, A.S.C.E., A.I.E.E., A.I.M.E., and A.S.M.E., died in Los Angeles, Calif., on May 25, 1935.

Mr. Buzby's work reached across the continent, from steel construction for Post & McCord in New York in 1912-1913 to consulting engineering for S. Charles Lee, architect, in Los Angeles, at the time of his death. He was bridge designer for the Erie Railroad Company from November, 1913, until the end of 1915, and in two periods of employment with the Chile Exploration Company and Braden Copper Company at New York, progressing from structural designer to principal assistant to the designing engineer, he designed structures for copper reduction works and smelters, prepared contract specifications for ore loading and unloading bridges, handled technical correspondence and reports, examined tenders for a wide variety of mechanical equipment, and designed conveyer systems and copper plants. He was engaged in these duties from 1915 until early 1920, with the exception of the months from April, 1918, to February, 1919, when he was experimental engineer for the Aviation Section, Bureau of Aircraft Production. He was attached to the French Aviation Mission to the United States and stationed at Detroit, Mich.

From February, 1920, to June, 1924, Mr. Buzby was technical consultant to the General Eastern Office of the Wellman-Seaver-Morgan Company, of Cleveland. His advice was sought on the de-

sign, layout, selection, and application of the mechanical equipment for coal and ore handling plants, hydroelectric plants, steel works, rubber plants, ocean freight and cargo handling terminals, and special shipbuilding and other types of large cranes. He left this company to become general manager of the Kinodyne Radio Corporation, New York, but suffered a breakdown in health in 1925 and was not active during that year. Subsequently he formed the firm of De-Saussure & Buzby and practiced consulting engineering in Jacksonville, Fla., from January, 1926, to July, 1932, when he closed that office and went to California, where the remainder of his life was spent.

Mr. Buzby was born at Montreal, Que., Can., on June 30, 1889. His parents, Winslow Edward and Clarine (Endall) Buzby, were United States citizens. He studied at Princeton University for three years, 1906-1908, and at Valparaiso University in 1910-1911, receiving a C.E. degree there. The following year he did postgraduate work at the Massachusetts Institute of Technology, specializing in structural and sanitary engineering, and secured an M.S. degree in 1912.

He became an associate member of the A.S.M.E. in 1919 and a member in 1924. He belonged to the Society of Terminal Engineers, in addition to the four national societies, and was a member of the Delta Kappa Epsilon fraternity. He won many honors in golf, and was interested in the collection of stamps. He is survived by his widow, Helena (Wood) Buzby, whom he married in 1919.

ERNEST MORRIS CLAYS (1905–1935)

Ernest Morris Clays, who was killed in an airplane crash at McGill, Nev., on April 10, 1935, was born at Bingham Canyon, Utah, on June 27, 1905, a son of William D. and Mary A. (Bennett) Clays. Since 1923, when they were attending the University of Nevada, he had been a close friend of William Keith Scott, now of the Scott Motor Company, Reno, who has provided the following record:

"In 1927 and 1928 Ernest and I spent a year traveling in Europe and North Africa. After returning to this country he went back to the university and completed his course, receiving a B.S. degree in mechanical engineering in 1931. After graduating he was with me both in Reno and Los Angeles. Part of the time he was assistant general manager of Scott Motors, Ltd., taking an active part in the credit end of the business. Whenever I had a business venture or a mine to be investigated he did the investigating for me. For a while he was at a mine that I was operating at Julian, Calif.

"At my instigation he went to the Boeing School of Aeronautics for a nine months' course in pilot training. At the end of six months he decided that it was no longer beneficial for him to stay there. He was always much interested in research engineering, and the main reason he went to the Boeing School was that we considered it an advantage for an aeronautical engineer to know how to fly. He wanted very much to get into aeronautical engineering, but the opportunity had not presented itself. There have been many things in the two years since his death that would have been most interesting to him, and we miss him greatly.

"The crash at McGill has never been satisfactorily explained. He was a very conservative pilot, and why he should get into a flat spin so close to the ground is unexplainable. He had flown the ship, which was a J-5 Stearman, a good many hours, and knew the ship. His brother, Leonard, died with him in the crash.

"He went through a full course of the Reserve Officers Training Corps at the university and was a lieutenant in the Army Reserve Corps.

"He was unmarried."

Mr. Clays became a junior member of the A.S.M.E. in 1931.

FREDERICK ARTHUR COLLINS (1893–1936)

Frederick Arthur Collins, engineer for the Union Gas & Electric Co., Cincinnati, Ohio, died in that city on January 4, 1936, after an illness of nearly six months.

Mr. Collins was born in Long Island City, N.Y., on November 24, 1893, son of John T. and Rosanna Collins. From the Flushing High School he entered Cornell University, which gave him an M.E. degree in 1915.

During the year after his graduation Mr. Collins was employed by the Remington Arms Company, of Bridgeport, Conn., as foreman of one of the production shops, machining small parts for a French rifle. He then secured a position with the Interborough Rapid Transit Company, New York, as assistant field engineer. He was first put in charge of the installation of turbines, piping, and boiler equipment at the 59th Street Power Station and upon the completion of this assignment became technical assistant to the mechanical engineer, handling all technical work in connection with the operation of plants throughout the city.

In 1924 Mr. Collins left the Interborough to become mechanical inspector for Murrie & Co., New York, and subsequently worked for the Griscom-Russell Company, New York, and for about three years as power engineer for the Texas Company in that city. He had been with the Union Gas & Electric Co. since November 1, 1928, serving as engineer with the Power Sales Department in connection with negotiations for industrial power contracts.

Mr. Collins became an associate member of the A.S.M.E. in 1924. He is survived by his father, his widow, the former Miss Loretta Knott, of Flushing, and by two brothers and two sisters.

JAMES VAN VORST COLWELL (1870–1937)

James Van Vorst Colwell, whose death occurred in New York on January 14, 1937, was born in that city on November 4, 1870. He was the son of Lizzie B. (Van Vorst) and Augustus W. Colwell, who was one of the charter members of the A.S.M.E.[1] Mr. Colwell became a member of the Society in 1903 and gave generously of his time to its activities. He served on the Special Committee on Increase of Membership, 1912–1916, and was always active in the development and work of the Metropolitan Section.

Mr. Colwell was a member of the Transportation Committee of the American Committee for the first World Power Conference, held in London in 1924, and he and Mrs. Colwell attended the conference. He also served on a special subcommittee on Opportunities of the Professional Engineers' Committee on Unemployment from 1930 to 1935.

Mr. Colwell attended public school in New York, Borden Military Institute of New Jersey, and Pratt Institute, Brooklyn, and also took courses through the International Correspondence Schools. He served an apprenticeship in the foundry and erecting shop of his father's ironworks from 1888 to 1891, and after some experience in the drafting room, engaged chiefly in the erection of sugar machinery at the shops until 1898. In that year he had charge of the erection and equipment of works for the U.S. Sugar Refining Company at Waukegan, Ill., and the following year inspected beet sugar plants in this country and cane sugar plants in Mexico. In 1900–1901 he was in charge of the construction, erection of machinery, and operation of a beet sugar plant for the Marine Sugar Company, Marine City, Mich. From there he went to Pottstown, Pa., where he supervised the erection of machinery for Stanley G. Flagg & Co. and the Keystone Foundry.

At this point Mr. Colwell's record shows a departure from sugar machinery work. He became superintendent of the C. W. Hunt Co., West New Brighton, S.I., N.Y., manufacturing hoisting and conveying machinery, and electric locomotives. He continued in that position until 1906, when he became associated in business with his sister's husband, Henry Harrison Supplee (Manager, A.S.M.E., 1897–1900), editor of *Cassier's Magazine*, as treasurer and manager of the publication. This arrangement was terminated in 1909, and Mr. Colwell took the position of manager of the New England Department of the Heine Safety Boiler Company, in which he continued until 1912. During the next few years he was connected with the Didier-March Company, manufacturers of firebrick of Perth Amboy, N.J., as sales engineer, and during the World War was engaged in the manufacture and sale of machinery for the production of shrapnel bullets. He was a member of the Seventh Regiment, New York, for eight years, and, subsequently, belonged to the Veterans Association.

After the War he carried on a private consulting business as equipment engineer for a few years, but was not active during the latter years of his life.

Mr. Colwell is survived by his widow, Elizabeth (Rodenbach) Colwell, whom he married in 1902.

ROBERT NEWELL CUNDALL (1875–1936)

Robert Newell Cundall, chief engineer of the Consolidated Packaging Machinery Corporation, Buffalo, N.Y., died on October 7, 1936, in New York, N.Y., of heart trouble. He was born at Putnam, Conn., on November 22, 1875, son of Charles L. and Jessie (Fremont) Cundall. He prepared for college in the schools of Putnam, and was graduated from Worcester Polytechnic Institute in 1897 with a B.S. degree in electrical engineering. From then until the latter part of 1901 he was connected with the sewer department of the City of Worcester, working on the design and installation of electrical and mechanical equipment. During the next two years he was master mechanic for the Bay State Pink Granite Company, Milford, Mass., and then for a period of equal length was employed in the master mechanic's department of the Draper Company, Hopedale, Mass.

In November, 1905, Mr. Cundall returned to his native state to

[1] Obituary published in Trans. A.S.M.E., vol. 39 (1917), p. 1207.

take a position as draftsman with the Yale & Towne Manufacturing Co., at Stamford. After six months he was promoted to assistant to the drawing office manager, and a year later became manager of that office. He went back to Worcester in December, 1908, and during the next year attempted to establish himself in professional work there. He had been located in Buffalo since that time, for the first three years with the Molyneux Mailing Machine Company, designing special automatic machinery, then in private practice. For a number of years, beginning in December, 1916, he was a partner in the firm of Cundall, Powell & Mosher, Inc., of which he was president, and then he became vice-president and chief engineer of the Industrial Planning Corporation. He had been with the Consolidated Packaging Machinery Corporation since December, 1927. He held several patents in this field.

Mr. Cundall became a member of the A.S.M.E. in 1918, and was also a member of the Buffalo Engineering Society. He was a Mason and Knight Templar. Surviving him are his widow, the former Luta Lincoln, of Worcester, whom he married in 1898, and two sons, Lincoln A. and Sanford H. Cundall.

FREDERICK STEWART CUTTING (1882–1937)

Frederick Stewart Cutting was born in Providence, R.I., on January 18, 1882, son of Robert and Julia H. (Mott) Cutting. Supplementing his common school education he took private instruction in mathematics and allied subjects. He began work at the age of sixteen and during the next six years had experience in the telegraph service, as a locomotive fireman, and at sea in engine and boiler rooms of ocean and coastwise steamers as oiler, fireman, watertender, and cadet engineer. In July, 1904, he was employed as assistant to the president and engineer of the Massie Wireless Telegraph Company, Providence. During the first part of his connection with this company, which continued through the year 1911, he was in charge of construction, testing, and development work, both commercial and for the Navy. In 1908 and 1909 he was engaged in power house and substation service for the New Haven railroad, then he returned to the construction and development of radio and wireless telephony towers and mechanical devices.

In 1912 Mr. Cutting worked for six months as fireman for the New Haven railroad. Subsequently he was employed by the New London Ship & Engine Co., at Groton, Conn., on the construction and testing of Diesel engines for submarines. He next became chief clerk of the Power and Line Department of the Rhode Island Company (serving local street railways), whose plant later became part of the New England Power Company.

Early in 1917 Mr. Cutting was at sea in the engine department of trans-Pacific ships burning oil. Shortly after the United States entered the World War he became an instructor in electrical engineering at the U.S. Navy Radio School at Harvard College. Later he served as Engineer Officer in the U.S. Navy, in connection with reconditioning former German liners at the Boston Navy Yard; as inspector of machinery for the U.S. Shipping Board, Emergency Fleet Corporation; and as assistant superintendent of the Lord Construction Company, at Providence, supervising the installation of machinery and equipment on a number of vessels outfitted at this plant. After the War he was again at sea on large oil-burning vessels, principally South American and West Indian.

From November, 1923, to October, 1926, Mr. Cutting was employed by the Dwight P. Robinson Co., of New York, and the Carleton-Mace Company, of Boston, on the construction of a 150,000-hp power plant for the New England Power Company. During intervals between the completion of contracts in this period he spent some time as engineer on the Colonial Line steamers, and on the Standard Oil tank ship, *Standard Arrow*. After the completion of the power plant he was for nine months appraisal engineer for the Jackson & Moreland Co., of Boston, on power house and public utility work.

For a short period in 1924 Mr. Cutting was employed in the office of the public service engineer at Providence, and in the fall of 1927 he was appointed deputy public service engineer for that city, and placed in charge of the smoke inspection department under the general direction of the public service engineer. He was appointed deputy public service engineer in May, 1929, and held that office for the remainder of his life.

Mr. Cutting held Government certificates as first assistant, ocean steamships of any gross tons; second assistant, Diesel ships of any gross tons; chief electrician in charge for naval radio; chief electrician for submarine vessels; qualified submarine diver on deep-sea work; and radio operator, commercial, first order.

He became a member of the A.S.M.E. in 1934 and belonged also to the American Institute of Electrical Engineers and the Providence Engineering Society. He had served as chairman of the library committee of the latter organization, and at the time of his death was chairman of its committee on employment. He was a Knight Templar and a Shriner.

Mr. Cutting died at his home in Providence on March 13, 1937. He is survived by his widow, Ida S. (Davis) Cutting, whom he married in 1905, and by one daughter, Gertrude D. Cutting.

CHARLES EARLE DAUGHERTY (1881–1936)

Charles Earle Daugherty, president, treasurer, and general manager of the Norbom Engineering Company, Darby, Pa., died at the Jefferson Hospital, in Philadelphia, on October 22, 1936. His widow, Anna Mary (Dunn) Daugherty, whom he married in 1911, survives him.

Mr. Daugherty was born at Thompsontown, Juniata County, Pa., on January 19, 1881, his parents being John Riley and Catherine (Fleet) Daugherty. He was valedictorian in the Class of 1899 at the Scranton High School and during the next year taught in public schools in Delaware Township, Juniata County. Then for three years he was connected with the International Correspondence Schools, of Scranton, dealing particularly with students of mathematics. He also taught in public night schools in Scranton in 1901. He was deputy prothonotary of Lackawanna County, Pennsylvania, from 1904 to 1907, and then, after serving again as teacher for the I.C.S. during the first part of 1907, became draftsman for the American Bridge Company, at Pencoyd, Pa. In 1908 he completed the course in mechanical engineering of the I.C.S. and entered the employment of James Boyd & Bros., Philadelphia. He was draftsman for a year and then chief draftsman until 1917, when he was made chief engineer.

All of his work for this company had to do with motor-driven fire apparatus and during the first six months of 1918 he was engaged in research work on liquid fire apparatus for the U.S. Ordnance Department as chief engineer of the Boyd Manufacturing Company, Philadelphia. From July to October of that year he was checker on airplane controls and machinery in the Engineering Department of the Naval Aircraft Factory at Philadelphia, then entered the Engineering Division of the Ordnance Department, Washington, D.C., where he served as first lieutenant until January 2, 1919, in research work in the Gas and Flame Section of the Division.

Following his release from service he took a position as chief engineer of the Norbom Engineering Company, manufacturers of hydraulic dredges and dredging machinery. He became vice-president, assistant treasurer, and general manager of the company in 1929 and its president and treasurer in 1935. He had patented a sealing device for a dredging or similar pump.

Mr. Daugherty's membership in the A.S.M.E. dated from 1929. He was an excellent pianist, and was assistant organist at the Methodist Episcopal Church in Wayne, Pa., where he made his home.

BRADFORD H. DIVINE (1869–1934)

Bradford H. Divine, president of the Divine Brothers Company, Utica, N.Y., died on November 24, 1934. He was born in Utica on July 5, 1869, and established the firm of Divine Brothers (afterwards incorporated under its present name) in 1894, for the manufacture of metal-finishing tools. The business expanded into the manufacture of metal-finishing equipment and the development of processes for the reduction and finishing of metals by flexible grinding and polishing. As president of the company Mr. Divine directed work for companies manufacturing a wide variety of products, from cold-rolled shafting to jewelry, and including saws, axes, gun and automobile parts, cutlery, and all kinds of tools. His knowledge of abrasives was extensive and he designed many tools and developed the processes for the work of the company.

Mr. Divine became an associate of the A.S.M.E. in 1917 and a member the following year. He had served as president of the Metal Finishing Equipment Association.

WILLIAM ALBERT EDWARD DOYING (1867–1936)

William A. E. Doying, inspecting engineer of the Panama Canal, died at his home in Washington, D.C., on August 3, 1936, after a prolonged illness. He had served in connection with the Panama Canal for more than 30 years.

Mr. Doying was born on June 13, 1867, at Danville, Ont., Can., son of Ira Edward and Sarah Jane Doying. His higher education was secured at Stevens Institute of Technology, for which he prepared at the Stevens High School. He attended the Institute for two years, with the Class of 1890, then went to work as a machinist in a molding and planing mill in New York.

During the years 1891–1896 he was located in Summit, N.J., where

he was a member of the firm of Doying Brothers, electrical contractors, for several years, and then superintended the remodeling of the Hotel Beechwood and the erection of cottages, the plans for which he had drawn; he also had charge of the installation of boilers, a hydraulic elevator, steam laundry machinery, gas machinery, and kitchen steam apparatus.

For a year, beginning in August, 1896, Mr. Doying was draftsman on telephone and switchboard work for the Western Electric Company, New York. During the next eight years he was engaged in general engineering work for the Metropolitan Street Railway Company, New York. He was squadmaster in connection with the construction of both the 96th and Kingsbridge power stations, checking drawings and contractors' prints for water, steam, exhaust, and circulating water piping, structural work, and similar details. For the substation and car house at Amsterdam and 129th Streets, the Bayard and Elizabeth Streets substation, and the Second Avenue and 65th Street substation, he was in charge as checker, laying out all electrical apparatus, rotaries, switchboards, and wiring. In addition to his work for the Metropolitan Street Railway Company, he assisted its chief engineer, F. S. Pearson, who was also consulting engineer for the Sao Paulo Tramway Light & Power Co., of Brazil, by checking plans and contractors' drawings for a 2000-hp hydroelectric generating station and stepdown station for that company.

In September, 1905, Mr. Doying was appointed assistant inspecting engineer for the Panama Railroad Company, with offices in New York. He was in charge of the inspection of all materials and machinery used in the construction of the Panama Canal and railroad. He was appointed inspecting engineer of the Isthmian Canal Commission and transferred to Washington in the fall of 1908. The Commission became known as The Panama Canal in 1914.

In connection with his position he was appointed a member of the Federal Specifications Board and served on numerous committees under the Board, such as the Electric Lamp Committee, Committee on Standard Cement Specifications, and Committee on Standardizing Rubber Specifications.

Mr. Doying was elected a member of the A.S.M.E. in 1907. He allowed his membership to lapse in 1924 but renewed it in 1929. He served as a member of the Executive Committee of the Washington Section in 1932. He was a member of the American Institute of Electrical Engineers, and also belonged to the Society of Automotive Engineers, Society of American Military Engineers, Washington Society of Engineers, and Cosmos Club, Washington.

He is survived by his widow, Caroline A. (Huttner) Doying, whom he married in 1900, and by two sons, Arthur C. and Howard H. Doying, all of Washington, D.C.

CHARLES WHITNEY DUNN (1865–1936)

Charles Whitney Dunn was born in Granville, Ill., on October 9, 1865, the son of James and Lusette Dunn. He was educated at the Granville High School, Oberlin Preparatory School, and Cornell University, receiving the M.E. degree from Cornell in 1893. From graduation until 1907 and from 1912 until 1928, with the exception of a few short intervals, he was with the Robert W. Hunt Company, Chicago and New York, engaged in testing, appraisal, and inspection work. This included inspection or appraisal responsibilities in connection with reciprocating and turbine power plants, railroad track construction, water-pumping machinery, locomotives, and steel buildings. During the War he was in charge of the inspection of materials ordered by the General Engineering Depot, Washington.

Ill health kept him from active work from 1907 to 1912 and from 1928 to 1930. In December, 1930, he became associated with the Polytechnic Institute of Brooklyn, designing and supervising the construction of equipment and apparatus in the metallographical, industrial, aerodynamic, and mechanical laboratories. He became interested in teaching and conducted some evening classes. During the interval from December, 1934, to June, 1936, he was with the Sperry Gyroscope Company, Brooklyn, on investigation and inspection work. He then returned to the Polytechnic Institute in connection with the design and construction of an experimental flume for the hydraulic laboratory. On September 8, 1936, he died suddenly from heart trouble, at the end of a day's work.

Mr. Dunn joined the A.S.M.E. in 1897. He was a member of the Municipal Club of Brooklyn and active in its work. He is survived by his wife, Ida (Conrad) Dunn, whom he married in 1895.—[Memorial prepared by Edwin F. Church, Jr., Brooklyn, N.Y. Mem. A.S.M.E.]

JAMES HERBERT EDDY (1909–1935)

James Herbert Eddy, who became a junior member of the A.S.M.E. in 1931, was born on February 26, 1909, at Hot Springs, Ark., the son of Frank Herbert and Olga (Helmsmiller) Eddy, and died on June 13, 1935, at Lynn, Mass. He entered the University of Arkansas from the Hot Springs High School and was graduated with a B.S. degree in mechanical engineering in 1930. This was supplemented by an advanced course in engineering at the General Electric Company, Schenectady, N.Y.

In 1931 and 1932, Mr. Eddy was employed in the testing department at the Schenectady works of the General Electric Company, and, for the next three years, was stabilizer engineer with the Bell Oil & Gas Co., Grandfield, Okla. In 1935, he returned to the General Electric Company, this time in the standardizing laboratory at the West Lynn, Mass., works, where he was engaged in developmental and experimental work.

Mr. Eddy was particularly interested in measuring and testing apparatus and welding apparatus and supplies.

GODFREY ENGEL (1860–1936)

Godfrey Engel, retired sugar-refining engineer, died on May 29, 1936, at the home of his daughter, Mrs. William V. Gosline, in Glens Falls, N.Y., where he had lived for two years. Surviving him is also his son, Godfrey Engel, Jr.

Mr. Engel was born at Zingst, Prussia, Germany, on July 6, 1860. He secured his technical education in the United States, attending the School of Mines of Columbia University for two years, 1876–1878.

He then served a three-year apprenticeship in the drafting room and shops of S. S. Hepworth & Co., New York, and continued in the employ of the company, superintending the erection of machinery at several sugar refineries in Brooklyn and Philadelphia in 1882, then designing and erecting machine works for the Hepworth Company at Yonkers.

From April, 1883, to August, 1886, he was chief engineer for the Brooklyn Sugar Refining Company, then returned to the Hepworth Company for about a year in the capacity of chief draftsman. Subsequently he was chief engineer for the Moller Sierck Sugar Refineries and chief draftsman for what was called the "Sugar Trust" of Brooklyn, for about a year each.

In 1889 Mr. Engel left Brooklyn to take the position of chief engineer of the Baltimore Sugar Refining Company, which he held until 1897. The Sugar Apparatus Department of Bartlett, Hayward & Co., Baltimore, then secured his services as chief engineer, and he continued in that position for about eight years. In 1905 he went to Pittsburgh, Pa., where for several years he was chief engineer of the Sugar Mill Department of the Riter, Conley Manufacturing Company.

In 1908 Mr. Engel became associated with the American Sugar Refining Company. He took charge of standardizing beet sugar plants, with his office at Fort Collins, Colo. Two years later he returned to Brooklyn, where he was chief engineer of design and construction of the company's refinery. He resigned in 1918 to carry on consulting and construction work under the firm name of the Godfrey Engel Sr. Construction Co. He had charge of the design of the Hershey Cane Sugar Mill, Cuba, and from 1918 to 1921 was chief engineer of the Sugar Mill Department of the Buffalo Foundry & Machine Co. Subsequently his work was restricted to consulting practice.

Mr. Engel became a member of the A.S.M.E. in 1892. He belonged also to the Masonic fraternity.

ARTHUR EMERY FREEMAN (1882–1935)

Arthur Emery Freeman died in Arlington, Mass., on October 15, 1935, after an illness of more than eight years. He had been a member of the A.S.M.E. since 1921.

Mr. Freeman was born on December 27, 1882, at Arlington, Mass., and was the son of Horace A. and Mary (Emery) Freeman. He prepared for college at the local high school and was graduated from the mechanical engineering course at Massachusetts Institute of Technology with an S.B. degree in 1905. From 1906 to 1909, he was employed by the Government as resident engineer in charge of preparation of plans for heating, ventilating, and plumbing equipment for the House and Senate Office Buildings, Washington, D.C., and supervised its installation. Subsequently he was appointed heating and ventilating engineer for the District of Columbia but resigned, after a few months, to engage in the practice of consulting heating and ventilating engineering at Toronto. In the nine years that Mr. Freeman was in Toronto, he prepared plans and specifications for and supervised the installation of heating, ventilating, and plumbing equipment for numerous schools, churches, hospitals, banks, and institutions. He prepared plans, in 1920 and 1921, for the mechanical equipment that was installed in the new office building of the John Hancock Mutual Life Insurance Company, Boston, Mass., under

the supervision of F. A. Waldron, consulting engineer for the owner. From 1921 to 1926, Mr. Freeman was connected with the firm of French & Hubbard, Boston, and, subsequently, was a consulting heating and ventilating engineer in that city, but had retired from business at the time of his death.

Between 1915 and 1920, Mr. Freeman invented two mechanical boat davits. The first of these was approved and used extensively by the U.S. Shipping Board and was credited with the saving of many lives during the World War. The other was exhibited at the 1920 Motorboat Show and was one of the only two exhibits mentioned by *The Scientific American* in its review.

His widow, Ethel (Davis) Freeman, whom he married in 1907, survives him.

GEORGE C. FRY (1878–1935)

George C. Fry died at the Joseph Price Memorial Hospital, Philadelphia, Pa., on May 5, 1935, of pulmonary embolism. He was born at Eberlys Mills, Pa., a village located about five miles southwest of Harrisburg, on November 2, 1878, his parents being Jonathan and Susan (Wilt) Fry. His education was acquired in public schools, from a private tutor, and at Drexel Institute, Philadelphia, Pa.

From 1899 to 1911, Mr. Fry was in charge of the design and construction of new power-plant equipment and its installation for the Howard W. Read Company, Philadelphia. He was general manager of the National Waste Company, a Philadelphia manufacturer of waste for packing railroad journal boxes and for wiping, for approximately four years. In 1916, he entered the employ of the American Laundry Machinery Company, Cincinnati, Ohio, as a sales engineer in its Philadelphia office. Mr. Fry was made manager of the Boston, Mass., office a year later and in 1925 was transferred to the New York, N.Y., office, where he was a sales engineer, specializing in hotel and institutional installations at the time of his death. In this capacity, he installed the laundry machinery in many leading hotels and hospitals in New York and the surrounding territory. Probably the most noteworthy of these were laundries for the Columbia Presbyterian Medical Center and Cornell University Medical College, both in New York, for which he not only designed the equipment but also supervised its installation.

Mr. Fry had been an associate-member of the A.S.M.E. since 1926 and also belonged to the Engineers Club of Philadelphia, the Kiwanis Club, New York, and the Order of 1606, Boston, and the Odd Fellows and Masonic fraternity.

He married Miss Ethel M. Hummell in 1905 and is survived by her and a son, William Rawlings Fry.

GEORGE GROSS FRYER (1854–1936)

George Gross Fryer, who died on January 28, 1936, was born on May 25, 1854, in Philadelphia, Pa., the son of George and Mary Jane (Smyth) Fryer. His application for membership in the A.S.M.E. in 1891 contains the following record of his education and work up to that time.

"My early education was obtained mostly in private schools. From the ages of ten to sixteen I pursued a course at the Protestant Episcopal Academy, Philadelphia. I served a regular apprenticeship in the machine business with John L. Knowlton, Sharon Hills, Pa., 1871–1875. After obtaining my majority I took a course in civil and mechanical engineering at the Polytechnic College of the State of Pennsylvania at Philadelphia, graduating in 1877 as bachelor of mechanical engineering. The degree of master of mechanical engineering was conferred upon me by the College in 1881.

"Shortly after graduating I accepted a position as chief engineer of the Delaware River Chemical Works, now Baugh & Sons Co., and remained with them until the spring of 1881. For nine years I had charge of the manufacturing department and the entire management of the estate of George Fryer, at the same time being connected with several enterprises of a mechanical and engineering nature.

"I am now first assistant to the chief engineer of the Solvay Process Company, Syracuse, N.Y., engaged in general engineering work connected with the manufacture of soda ash, caustic soda, and bicarbonate of soda."

In a record given the Society in 1924, there is also a reference to studies at The Franklin Institute in 1871–1873, while Mr. Fryer was serving his apprenticeship, and a note indicating that the management of his father's estate entailed the design and manufacture of art furniture.

This later record shows that Mr. Fryer was purchasing agent for the Semet-Solvay Company from 1900 to 1908, estimating engineer for the Solvay Process Company from then until 1913, and consulting engineer for the company during the next five years. He was also secretary-treasurer and manager of the Salina Realty-Security Company for many years, beginning in 1903, and of the Lux Development Company from 1915 to 1918. He was secretary and manager of the Globe Graphite Company from 1916 to 1918.

Mr. Fryer was for five years an associate member in Company A, First Regiment, National Guard of Pennsylvania. He was a member of the Engineering Committee for Onondaga County, U.S. Fuel Administration, in 1917–1918. He has served as president and director of the Onondaga Historical Association and Syracuse Museum of Fine Arts, and was president of the Tech Club, Syracuse, for four years. He was a trustee and member of the Building Committee of the First English Lutheran Church of Syracuse.

Mr. Fryer married Regina Fremmel, of Syracuse, in 1901. Both she and their son, George von Albade Fryer, predeceased Mr. Fryer.

EDWIN W. GOESER (1882–1936)

Edwin W. Goeser was born on February 20, 1882, in Cincinnati, Ohio, son of August and Emilie (Kaiser) Goeser. He secured his education in his native city, and after graduation from a technical school there in 1899 went to Seneca Falls, N.Y., where he was employed for a year as draftsman for the American Fire Engine Company. He then returned to Cincinnati and from July, 1900, to October, 1905, was draftsman and designer for the R. K. Le Blond Machine Tool Co., working on lathes, milling machines, and special machinery and tools.

During the remainder of his life Mr. Goeser was located in California. He was associated with R. S. Masson, consulting electrical engineer of Los Angeles, from November, 1905, to June, 1906, designing miscellaneous electrical appliances, following which he spent brief periods as chief draftsman on small gas engines and special machinery for the McCan Mechanical Works in that city and in the Mechanical Department of the Los Angeles Pacific R.R. Co., at Sherman.

In April, 1907, he began a long engagement with the Union Tool Company, of Torrance. In his capacity as chief engineer of this company he was in charge of the engineering department and directed the manufacture of oil-well tools and equipment for both standard and rotary drilling, pumping equipment, gas, steam drilling, and crude-oil engines, steam and rotary pumps, and tractors. In 1923 he became associated with E. M. Smith, being made general manager of the Emsco Tool Company, Los Angeles, and when that company was merged with the Emsco Steel Products Company in 1926 to form the Emsco Derrick & Equipment Co., Mr. Goeser was made second vice-president and manager of production, the position he held at the time of his death on January 21, 1936.

The following paragraph regarding his special inventions and designs is taken from an obituary published by *Petroleum World*, in its February, 1936, issue:

"While it is not as widely known as his work in the heavy-duty equipment field, Mr. Goeser developed one of the first electrical vacuum cleaners, which he sold to the Hoover Vacuum Cleaner Company in 1911. He is better known in the petroleum-equipment field for having designed the first California-built rotary machine, which embodied outstanding improvement in its enlarged design and construction. Heavy-duty drilling equipment and the evolutionary trend in rotary drilling machinery since rotary machines first were tried in California, were engineered almost entirely by him."

Mr. Goeser had been a member of the A.S.M.E. since 1917. He resigned as a director of The American Petroleum Equipment Suppliers Association in November, 1935. He was a member of the California Oil and Gas Association and belonged to the Masonic fraternity. He played the violin and was particularly devoted to tennis. Surviving him are his widow, Florence M. Goeser, a daughter, Janet E. Goeser, his mother, and a sister.

JAMES BALE GREEN (1878–1935)

James Bale Green, the first employee to be hired by the Lucas Machine Tool Company, Cleveland, Ohio, when it started business in 1900, and its superintendent of production at the time of his death on November 8, 1935, was a native of Canada. He was born at Hamilton, Ont., on May 8, 1878, son of Thomas Allen and Mary (Bale) Green. His education was acquired in the public schools of Cleveland and through International Correspondence Schools courses in mechanical engineering and designing.

Mr. Green served as an apprentice in the shop and drafting room of the Warner & Swasey Co., Cleveland, from 1895 to 1899, and, upon completion of his apprenticeship, was employed by the Westinghouse Machine Company for approximately one year. When the Lucas Machine Tool Company began operations in 1900, he was engaged as a draftsman and remained with that organization until he died. He served successively as designer, chief draftsman, and superinten-

dent of production and, in these capacities, contributed many improvements to the company's products.

In addition to the A.S.M.E., of which he became a member in 1912, Mr. Green belonged to the American Society for Metals and was active in the Associated Industries of Cleveland. In 1905, he joined the Civil Engineers' Club of Cleveland, now the Cleveland Engineering Society, and was elected a veteran member in 1931. Mr. Green was a Knight Templar and a Shriner. His particular interests were his family, his garden, and anything, either large or small, that pertained to boring machines. He is survived by his widow, Lillian E. (Broome) Green, whom he married in 1900, a daughter, Ruth M. Rash, and a son, Donald C. Green.

JAMES ALEXANDER HALL (1888–1936)

James Alexander Hall, for many years active in the affairs of the A.S.M.E. and a vice-president elect of the Society, died on October 29, 1936. He was born in Berlin, Vt., on July 26, 1888, son of Joseph and Agnes Brock (Hardie) Hall. He attended the public schools of Providence and was graduated from Brown University in 1908 with the degree of A.B. He received the degree of Bachelor of Science in mechanical engineering from the same institution in 1910, remaining as assistant and then instructor in mechanical engineering until 1914, when he became connected with the engineering department of the Link-Belt Company in Philadelphia, Pa. The following year he returned to Brown University as assistant professor of mechanical engineering. He became associate professor in 1920, and in 1925 was appointed professor of mechanical engineering. Professor Hall also carried on a consulting engineering practice, and was associated with the Brown & Sharpe Manufacturing Co. in Providence, R.I., as consulting engineer from 1926 until the time of his death. In 1919 Professor Hall married Miss Leila Tucker, who with three children, a son and two daughters, survives him. He was a member of the University Club of Providence, the Faculty Club of Brown University, and of the Central Congregational Church of Providence.

Professor Hall joined the A.S.M.E. as a junior in 1912, became an associate member in 1918, and a member in 1923. At the Council Meeting in December, 1936, he was posthumously elected the first Fellow of the Society. He was the first chairman of the Providence Section, serving for two years, 1921–1922, and a member of the Standing Committee on Local Sections from 1922 to 1926, being chairman the last year. He had been honorary chairman of the Student Branch at Brown University since 1928. He was a member of the Research Committee on the Cutting and Forming of Metals from 1923 to 1931, and chairman in 1926 and 1927. He was chairman of the Nominating Committee of the Society in 1929, and a member of the Committee on Constitution and By-Laws, 1931 to 1933. He was elected a manager of the A.S.M.E. in 1933 and served on the Executive Committee of the Council from then until his death. At the time of his death he was also chairman of the Machine Design Committee of the Machine Shop Practice Division and a representative of the Society on the Joint Committee on the Unwin Memorial.

Professor Hall was a past-president and a very active member of the Providence Engineering Society and was a past-chairman of the New England Section of the Society for the Promotion of Engineering Education, a member of the Newcomen Society and the Society of Sigma Xi, and a fellow of the American Association for the Advancement of Science.

He was one of four authors of "Profitable Science in Industry," published in 1924 by Macmillan. He also contributed many technical articles to the engineering journals and the publications of the A.S.M.E. He taught courses at Brown University in mechanism and machine design, applied mechanics, and industrial management.

When the State Employment Service decided in 1935 to conduct competitive examinations similar to those of the Civil Service, Professor Hall was appointed special representative of the United States Employment Service to conduct the tests.

While in college he was an outstanding student, winning, among other honors, the first Hartshorn premium in entrance mathematics in his freshman year. As a teacher he had a mind which penetrated to the very core of a problem and an unusual capacity for making the intricate and obscure clear and intelligible to others, qualities which inspired confidence and kindled enthusiasm in his students. He was recognized nationally as an outstanding consultant in the field of applied mechanics and design, and his irresistibly likable qualities won for him a host of friends. He was the type of man whom one naturally addressed by his first name. As an indication of the qualities which made him so universally admired, the following quotation from a letter written to Mrs. Hall by Harry R. Westcott and Professor Samuel W. Dudley on behalf of the Regional Group I Delegates Conference of the A.S.M.E. is of interest:

"We knew Jim Hall intimately as a fellow worker, companion, and friend. In all these relations, we admired his steadfast devotion to a high idealism; we delighted in the vivid play of his always kindly humor, and we were warmed by the sincerity of his inclusive friendliness.

"He was a resourceful and stimulating contributor to our meetings, ever ready to give freely of his time and abilities and always effective in elevating the professional level of our discussions and in promoting the friendly coöperation of our members.

"We are deeply indebted to him for his outstanding services to the Local Sections of the Society, and particularly to those here in New England, both as an official in our National Society and as a fellow member. His keen insight and wide personal experience in Society affairs enabled him to bring to our meetings and conferences information and interpretations of actions and trends which quickened our understanding and deepened our appreciation. He renewed and strengthened our loyalty to the Society of which he was so proud and to which he devoted himself so unsparingly and so effectively.

"In the shadow of our great loss, we are uplifted by the memory of his enthusiasms, his zeal, and his rare qualities of mind and heart. We are profoundly grateful for his influence upon the host of young engineers who were guided by his counsel and inspired by his example. These, his living monuments, through many years and in many lands, shall rise up and call him blessed."—[Memorial prepared by W. H. Kenerson, Providence, R.I. Mem. A.S.M.E.]

RICHARD HAWORTH (1896–1935)

Richard Haworth was born at Dorchester, Mass., on February 10, 1896, son of Samuel and Emma (Smethurst) Haworth. He attended the Mechanic Arts High School in Boston and entered Tufts College with the Class of 1919. In 1917 he enlisted in the naval air force, in which he held the rank of lieutenant when discharged in 1920. He returned to take his B.S. degree in engineering at Tufts in 1921, and since that time had been connected with the Otis Elevator Company.

He spent his first year with the company in the factories, studying methods of manufacture, installation, and maintenance. He was then put in charge of the office located at Huntington, W.Va., covering the territory in southern West Virgina and eastern Kentucky. Then he was transferred to Indianapolis, covering the State of Indiana, and next was made local manager, at Baltimore, for the State of Maryland. Subsequently he was made manager of the Baltimore Zone of the company, comprising the territory covered by the States of Maryland, Virginia, and North Carolina, and the District of Columbia, and was serving in that capacity at the time of his death on May 14, 1935.

Mr. Haworth became an associate of the A.S.M.E. in 1927. He belonged to the University Club of Indianapolis, the Maryland Club of Baltimore, and the Baltimore Country Club, and was a member of Delta Tau Delta fraternity and a Mason.

Surviving Mr. Haworth are his widow, Louise (Chester) Haworth, whom he married in 1928, and two children, Joan Louise and Cathleen Chester Haworth.

GEORGE ERNEST HEATH (1881–1935)

George Ernest Heath, chairman and managing director of Carels Diesel & Steam Engines Ltd., London, England, since 1919, died on March 18, 1935. He was born at Cottingham, East Riding, Yorkshire, England, on July 3, 1881, and was educated through private tutoring and at Hull Technical College. His first position was that of assistant engineer for the York & Dundee Municipal Electrical Works, in 1902–1903. During the next seven years he was engineer and manager for Erith Council Electric Light & Transportation, in London. He was engineer for the Diesel Engine Co. Ltd., London, from 1910 to 1914, and from then until the close of the World War was chief engineer and manager of the Chilwell National Shell Filling Factory, directing the filling of high-explosive shells.

Mr. Heath became a member of the A.S.M.E. in 1929 and also belonged to the Diesel Engine Users Association.

WILLARD HENRY HERR (1886–1937)

Willard Henry Herr, sales engineer in charge of the New York office of the Hydraulic Press Manufacturing Company, of Mt. Gilead, Ohio, died at his home in Leonia, N.J., on April 11, 1937. He is survived by his parents, William Henry and Carrie Estella (Root) Herr, a sister, Mrs. James De Witt Patten, of Windber, Pa., and by his widow, Anna Elizabeth (Jones) Herr, whom he married in 1917. He had been a member of the A.S.M.E. since 1923.

Mr. Herr was born on October 23, 1886, at Columbia, Lancaster County, Pa., and attended school there. He served an apprentice-

ship as machinist and draftsman with the Wilson Laundry Machinery Company, of Columbia, and worked for a year in their drafting room. He then went to Pittsburgh where he was employed for about a year as machinist and draftsman by the Dau-Marsch Company, working on the design of an experimental rotary gasoline engine. Following this he spent something over two years with the Pullman Motor Car Company, York, Pa., designing tools and jigs for manufacturing automobile engines.

In October, 1910, Mr. Herr took a position with the Columbia Manufacturing Company and until March, 1912, was in responsible charge of the design and installation of steam laundry machinery for special requirements. During the next two and a half years he was designing draftsman for the Westinghouse Machine Company, East Pittsburgh, working on horizontal gas engines and machine tools. Then after a few months with the Lake Erie & Ohio River Canal Board, Pittsburgh, as draftsman and computer, he became engineering salesman and designing draftsman for the Hydraulic Press Manufacturing Company. His first engagement with the company was only for a few months, after which he was recalled by the Westinghouse Machine Company to work on the design of steam turbines for central power station and marine use. He became manager of the New York office of the Hydraulic Press Manufacturing Company in August, 1918.

THEODORE HENRY HINCHMAN (1869–1936)

Theodore Henry Hinchman, the son of John Marshall Hinchman and Ella (Cropsey) Hinchman, was born in Detroit, Mich., on June, 24, 1869. His forebears were important people in Detroit back in the days when it began to develop from a town almost exclusively French in character, into an American city. His great-grandfather, Dr. Marshall Chapin, was one of Detroit's early physicians, having left his home in New York State in 1809 to settle there. In 1819, his grandfather, Theodore Henry Hinchman, established a wholesale drug business, which, after the founder's death, was carried on by his sons under the name of T. H. Hinchman's Sons, and which ultimately became The Michigan Drug Company of the present time. With such a background, the subject of this memoir may well be called a "Detroiter," since he and his relatives have taken an active part in the growth and development of the city for more than one hundred and twenty-five years.

As a boy, Mr. Hinchman attended the Detroit Public Schools, and was graduated from the Old Detroit High School in 1887. He then entered the University of Michigan, at Ann Arbor, Mich., from which he was graduated from the Liberal Arts Department in 1891, with the degree of Bachelor of Arts, and from the Engineering Department, in 1893, with the degree of Bachelor of Science. While acquiring his training as an engineer, he had the good fortune to study under Mortimer E. Cooley, beginning a friendship which continued for the remainder of his life.

During college vacations he worked on the engineering staff of the Michigan Central Railroad Company, and under Henry M. Leland, with the Leland, Faulkner, Norton Company. After his graduation, in 1893 and for a part of 1894, he served under Professor Cooley and under Jesse M. Smith, on consulting engineering work.

In the latter part of 1894, the partnership of Field and Hinchman, consulting mechanical and electrical engineers, was formed. Times were hard in those years immediately after the panic of 1893; and it took real courage to start a new business, venturing all one's small financial resources. Fifteen hundred dollars was the paid-in capital with which the new firm started. Of this amount, Theodore Hinchman contributed $1000 which had come to him through a legacy from his maternal grandfather, John G. Cropsey, of Chittenango, N.Y. George H. Field had also been recently graduated from the University of Michigan.

As can be seen by the dates, this was long before the development of the automobile industry in Michigan. Detroit was a slow, easy-going city of about 250,000 people—approximately one-sixth of its present size. It was a pleasant, clean place in which to live, being largely a jobbing center with comparatively little manufacturing. The first steel-frame office building was constructed about this time. There was very little work for the consulting engineer in such a community, and it was very difficult to secure enough of it "to keep the wolf from the door."

A year later, on October 24, 1895, Theodore Hinchman married Emma MacAllan Ballentine, at her home in Port Huron, Mich.

When war was declared against Spain in 1898, Mr. Hinchman temporarily left his work in Detroit to enlist in the Navy as chief machinist on the U.S.S. *Yosemite*, serving under his old friend and professor, Mortimer E. Cooley, who was chief engineer of that ship. The *Yosemite* served in the blockades of Santiago, Cuba, and of San Juan, Puerto Rico.

After the war, Mr. Hinchman resumed his work as consulting engineer; and by hard, faithful, and effective service built up his reputation.

The advent of the automobile business, and the increase in manufacturing in Detroit after 1900, brought more demand for the services of a man who had proved his ability and reliability during the preceding years. There was a great increase in the use of structural steel, and reinforced-concrete construction was introduced. Building work required the services of the engineer as well as those of the architect. On March 30, 1903, an Architectural Department was added to the firm's activities; and the business was incorporated under the name of Field, Hinchman, and Smith.

In an old city like Detroit, many of the downtown office and commercial buildings occupied the best sites, and were hopelessly inadequate for the requirements. Obviously, they would soon have to come down, and be replaced by modern structures. New factories, embodying up-to-date manufacturing methods of production, were constantly going up. In planning the construction and equipment of such buildings, the harmonious and cooperative work of engineers and architects was necessary.

At the end of 1906, Mr. Field left the organization; and the present name—Smith, Hinchman and Grylls, Inc.—was adopted early in 1907.

The following are some of the important pieces of work planned and carried to completion under Mr. Hinchman's supervision: Buhl Building (26 stories); Union Guardian Building (34 stories); Penobscot Building (44 stories); J. L. Hudson Co. Department Store; Crowley Milner Co. Department Store; Second National Bank of Saginaw, Mich.; the University Club (Detroit); the Detroit Country Club; Dodge Brothers Automobile Plant; Hiram Walker & Company's great distillery recently completed in Peoria, Ill.; the Mistersky power plant and transmission system for the Public Lighting Commission of Detroit; the high-pressure fire protection system of Detroit; the Fairview sewage pumping plant for Detroit; and the power plant, the Chemistry Building, substation, power house, East Engineering Building, Yost Field House, Intramural Building, and Graduate School, of the University of Michigan.

In 1905, Mayor George P. Codd recognized the ability of Theodore Hinchman, and the need the City of Detroit had for his services, by appointing him a member of the Public Lighting Commission. He was also a member of the Michigan State Planning Commission, receiving his appointment from Governor Fitzgerald, in January, 1935. At the time of his death, he was president of the Village of Grosse Pointe Farms, where he had his home.

Soon after starting in business in 1894, Mr. Hinchman assisted in organizing the Detroit Engineering Society. For several years he served as secretary of this society, holding this office until it had become firmly established as a factor in the Detroit community. Later, he became, successively, vice-president and president of the society. His affiliations also included membership in the Detroit Board of Commerce, of which he was at one time a director, the Detroit Country Club, the University Club, and the Detroit Club, of which a few years ago, he was president.

Coming from an old Presbyterian family, Mr. Hinchman was always a member and consistent worker in that church and, at the time of death, he was president of the Board of Trustees of the Grosse Pointe Memorial Presbyterian Church.

Theodore Hinchman is survived by Mrs. Hinchman and their three sons, Theodore Henry Hinchman, Jr., David Ballentine Hinchman, and John Marshall Hinchman, II. The second son, David, is married and has two children, one of whom bears the name of Theodore Henry Hinchman. He was a loving and much-loved husband and father.

In writing this memoir, the attempt has been made very briefly to sketch the career of one who represented the highest type of American citizen. He was fortunate in having back of him an ancestry that stood for the best things in life—integrity, reliability, and honor. He was endowed with an exceptionally bright and logical mind, and the capacity for unflagging industry, so that year by year he became more useful and more efficient. His education only began in his school and college days. It ended only with his death, which occurred at his home in Grosse Pointe Farms, a suburb of Detroit, on July 16, 1936.

Mr. Hinchman had a cheerful, friendly disposition that inspired in all with whom he came in contact, feelings of confidence, trust, and loyalty. He began his professional career at the "bottom of the ladder," relying for success on his own ability and industry. By consistent hard work, he progressed until he built up one of the finest engineering and architectural organizations in the United States.

When his services were needed by his friends, his church, his city, his state, or his country, they were freely given; he never shirked his responsibilities. It is an honor to have been his friend.

Mr. Hinchman became a junior member of the A.S.M.E. in 1895 and a member ten years later. He was also a member of the Ameri-

can Society of Civil Engineers.—[From a memoir prepared by William R. Kales, Detroit, Mich., for the Transactions of the American Society of Civil Engineers.]

DAVID EDGAR HUNTER (1861–1935)

David Edgar Hunter, chief mechanical engineer of the Shaw-Walker Company, Muskegon, Mich., died on November 15, 1935. He was born at Alberton, P.E.I., Canada, on October 26, 1861, son of David and Flora (Arbuckle) Hunter. During his attendance at the local grammar school, from which he was graduated in 1878, his spare time was spent in his father's shop doing manual training work. From 1878 to 1880, he continued his manual training in forging, machine-shop practice, and woodworking and was in charge of a small steam plant for approximately one year.

In 1880, Mr. Hunter emigrated to the United States and, for the next six years, worked for various builders in Beverly and Salem, Mass. During this time, he engaged in special studies on building construction and architecture. From 1886 to 1888, he was interested in the bicycle business at Salem. The next two years, Mr. Hunter worked for a manufacturer of interior trim and cabinet work at Boston, Mass., and was shop foreman when the Library Bureau acquired the business in 1890. He was appointed shop superintendent by the Library Bureau in 1891 and, during the four years that he held this position, studied machine design and architecture. Promotion to mechanical expert in 1895 brought new duties in connection with the design of new products, development of machines for their manufacture, and factory development and organization. His work from that time until 1911 was of a special creative character and included the design and construction of automatic machines for the production of index cards and similar filing devices, invention of the vertical file unit, development of steel furniture for offices and steel equipment for libraries and public buildings, and the design, construction, and installation of book stacks in the New York and St. Louis Public Libraries.

In 1911 and 1912, Mr. Hunter was employed by McKim, Mead & White, New York, N.Y., and had charge of the construction of the metal interior trim and furniture that was installed in the new Municipal Building of the City of New York which was constructed under their supervision. Upon the completion of this contract, he was engaged in reorganization work for the Empire Cream Separator Company, Bloomfield, N.J., in 1912 and 1913. Mr. Hunter was then employed by the Shaw-Walker Company as a specialist in steel furniture construction and, in collaboration with L. C. Walker, president of the company, organized and developed a steel furniture department. He continued with this organization as designing engineer and chief mechanical engineer and was actively engaged in these duties until two days before his death on November 15, 1935. In addition to his work as an engineer, which involved the creation of numerous new ideas in steel furniture construction, Mr. Hunter was a member of the board of directors for the last 15 years of his life.

Mr. Hunter was a prolific inventor, and several hundred patents were issued to him. One of these was for an improvement in combination press dies and was issued about 1880; another for a spring bicycle fork was issued several years later and was sold to the Pope Manufacturing Company, Hartford, Conn., who used it on safety bicycles until the introduction of pneumatic tires in 1891; others covered the vertical file unit now so generally used in business establishments, automatic machines for large-scale production of index cards, library book stacks, office furniture, and improvements thereon. He was generally recognized as the foremost authority on steel furniture construction in the United States and received a medal at the San Francisco Exposition in 1915 for meritorious work in the office-equipment field.

On November 26, 1887, Mr. Hunter married Mary C. Shields, who survived her husband by slightly less than a year and died on October 21, 1936. A daughter, Mrs. Fred Innes Brown, of East Greenwich, R.I., survives them. Mr. Hunter was primarily interested in his home and was a great student and reader.

Mr. Hunter was elected a member of the A.S.M.E. in 1916.

CLIFFORD MACKAY HUSTED (1886–1936)

Clifford Mackay Husted, who died at his home in Worcester, Mass., on September 29, 1936, was not only a skilled exponent of his profession, but had also the qualifications of an able organizer and administrator. He could plan and he could complete a project, taking care of all the multitudinous details that inevitably encumbered progress from the beginning to the end. He gave distinguished service during the World War at a most difficult post of duty. Not many months before his death he had been called upon to administer the activities occasioned by one of the most disastrous floods in the history of Massachusetts.

Major Husted was born in Buffalo, N.Y., on October 7, 1886, son of Edwin Merton and Flora (Mackay) Husted. He attended the Buffalo schools and the Cornell University School of Engineering, from which he was graduated with an M.E. degree in 1909. During the next five years he worked with his father as plant superintendent of the Husted Milling Company, Buffalo. He supervised the layout and construction of mill additions and the installation of machinery, and directed the changing of the entire plant from steam to electric drive. In 1914, when the milling business was discontinued and the company turned to elevator work, Mr. Husted entered the employ of the Standard Oil Company at its Eagle Works in Jersey City. He was soon assistant superintendent of the works and continued in much the same sort of duties as he had performed at Buffalo.

When the United States entered the World War in 1917, he volunteered his services and was assigned to the 104th Engineers, 29th Division, with the rank of captain. In 1918 he was advanced to major in the Engineers Corps, and later in the year was transferred to the Quartermasters Corps, and served at Sea Girt, N.J., and at Norfolk and Newport News, in Virginia. Particularly important was his assignment at the Port of Embarkation at Newport News. Here he handled the supplies, routing, and preembarkment training of thousands who sailed for France through Hampton Roads. He served from August, 1917, to December, 1918, subsequently becoming major in the Ordnance Reserve Corps.

Following his war service, Major Husted returned to the employ of the Standard Oil Company, becoming manager of the Refining Department of the Humble Oil & Refining Co., at Houston, Tex. He built the Baytown Refinery, at a cost of approximately $20,000,000, and operated it and several additional refineries and gas plants in Texas and Oklahoma in the years 1919–1921. The Standard Oil Company then sent him abroad to inspect its European plants from an operating point of view. Upon his return he was made general superintendent of the Eagle Works, where he remained until 1926.

During the next few years Major Husted was located in New York, N.Y., part of the time as president of the Combustion Service Company and later with the firm of Pask and Walbridge. Since going to Worcester he had served as president of the Boston Pressed Metal Company and vice-president of the E. D. Ward Company, builders and contractors.

In the fall of 1934, Major Husted was appointed by Joseph P. Carney, Massachusetts administrator of the then ERA, to go to Texas to study slaughtering and canning in connection with a project to ship into Massachusetts thirty to forty thousand cattle, menaced by a drought. The plan was to fatten the animals in pastures there, slaughter them, can the meat, and distribute it to the needy. Only a few hundred cattle reached New England pastures, however, before the ERA was supplanted by the WPA and the project was dropped. He was also connected with the Federal Housing Administration.

The organization and administration work in the Red Cross Society by Major Husted was especially notable. During his seven years' residence in Worcester he had become a recognized leader in many phases of the society's activities. Not long before his death he had sponsored the introduction and multiplication of first-aid stations along the highways, and had been chosen vice-chairman of the Highway First Aid, for the Worcester Chapter. As a staff assistant he was in the midst of a lecture series on first aid when death ended his work.

He came prominently into the foreground of Massachusetts affairs in the spring of 1936, when floods assumed disaster proportions. Realizing before others the necessity for immediate action, he proffered his services and engineering experience to Dr. E. L. Hunt, then Worcester chairman of the Red Cross. Major Husted was made superintendent of the quickly organized warehouse and disbursing system, Worcester. Commanding scores of Junior League members, American Legionnaires and their auxiliaries, and Boy Scouts, as well as members of the Red Cross, Major Husted managed the receiving and disbursing of some one hundred tons of donated clothing, bedding, furniture, and food. This was hurried by donated trucks to the hardest hit of the Massachusetts flood area, Hadley and Springfield. Dr. Hunt paid high tribute to Major Husted, to his engineering and administrative experience, and to the efficient operation of his system during the flood and in the highway first-aid stations.

Major Husted became a junior member of the A.S.M.E. in 1910 and an associate-member six years later. He was automaticallly transferred to the grade of member in 1935. He was a member of the Worcester Post No. 5, American Legion, the Military Order of the World War, the Officers Reserve Training Association, and the Army Ordnance Association. Fraternally he was affiliated with Sigma Alpha Epsilon.

In 1922 Major Husted married Arleen Hackett and he is survived by her and by their son, Chester C. Husted, of Waterbury, Conn., and also by his mother and a brother, Paul H. Husted, of Buffalo.—

[Based in part on a biography compiled by The American Historical Society for the "Encyclopedia of American Biography."]

WALTER WELDON JACKSON (1870–1935)

Walter Weldon Jackson was born in East Orange, N.J., on January 4, 1870, a son of Francis Whiting and Adeline (Egbert) Jackson. He received a degree in mechanical engineering from Stevens Institute of Technology in 1889 and after a short period of employment with the Safety Car Heating & Lighting Company, in New York, went to Providence, R.I., to work as draftsman for the Builders Iron Foundry. From the drafting room he went to the machine shop and later was put in charge of the venturi meter department.

At the Builders Iron Foundry Mr. Jackson was associated with Frederick N. Connet,[1] a classmate at Stevens, who had gone to Providence immediately following his graduation. They collaborated in the invention of the first registering device for use with Herschel's venturi tube. It was completed in 1892 and a 36-in. tube with the new register was exhibited at the Chicago World's Fair in 1893, measuring all water supplied by the city. The invention brought the award of the John Scott Medal to Connet and Jackson in 1898.

Mr. Jackson left the Builders Iron Foundry in May, 1899, to join the Providence Engineering Works as assistant superindendent, but resigned in March of the following year to accept the position of superintendent of the Wheeler Condenser & Engineering Co., in Carteret, N.J.

Early in 1904 Mr. Jackson took up consulting work. He was called to Columbus, Ohio, in connection with the design of water works there and remained to direct the construction of a storage dam, filtration plant, and sewage plant, and to supervise the water works system for about eight years. After his return East, he was for some years connected with the Regina Company, Rahway, N.J., as chief draftsman and superintendent, and carried on extensive consulting practice in the design and construction of municipal water works and sewage disposal plants. In 1899 he had patented a controller for filters, designed to maintain a uniform rate of discharge from a filter unit under varying conditions of the bed, and this was followed by other chemical feed regulators and filter controllers which greatly increased the scientific operation and economy of water works plants. For some years prior to his death he was engaged in the manufacture of tools and small parts of machinery.

Mr. Jackson became a member of the A.S.M.E. in 1902 and also belonged to the Chi Psi fraternity. He married Ellen W. Halton in 1894 and is survived by her and their daughter, Frances Halton Jackson, as well as by four sisters and a brother. His death occurred on January 1, 1935, at his home in Rahway.

WILLIAM BENJAMIN JACKSON (1870–1937)

This distinguished member of a distinguished family of engineers, third son of Prof. Josiah Jackson and Mary Detwiller (Price) Jackson, was born in Kennett Square, Pa., June 23, 1870. His father was professor of mathematics in The Pennsylvania State College, where Jackson received his preparatory and collegiate education, graduating in 1890, with the degree of Bachelor of Science, in the mechanical engineering course. Five years later he was awarded the degree of Mechanical Engineer for his accomplishments in practical experience.

Although educated as an engineer, his first employment was in the El Paso National Bank at Colorado Springs, where he remained for three years before returning to the engineering field, in which his work thereafter was almost wholly in affairs relating to electrical machinery and electrical power.

A rolling stone gathers no moss, but by rolling it gathers experience than which nothing is of greater value to a young engineer, and so it proved in the case of young Jackson. He resigned his position with the Colorado Springs Bank in 1893, to take charge of the Pennsylvania mining exhibit in the Chicago World's Fair Exposition. During the eight years following the close of the Exposition he held important engineering and supervisory positions at home and abroad with eight different manufacturing and power companies, and then with this wealth of experience joined with his brother in the formation of the firm of D. C. and Wm. B. Jackson, consulting engineers.

At first the office of the Jackson firm was located in Madison, Wis., but in 1907, better to handle the firm's growing business, the office was moved to Chicago. Wm. B. Jackson was placed in charge of the Chicago office and he made his home in that city until the dissolution of the firm in 1918. During this interval Mr. Jackson interested himself in the affairs of the Western Society of Engineers and served as its president during the year 1915. At the annual meeting of the society in January, 1916, Past-President John W. Alford tendered the society a fund for the establishment of an "Honor Award" and suggested the appointment of a committee to draft and submit a recommendation for its administration. With Mr. Jackson as chairman, the committee drafted a report recommending the name "Washington Award" and a set of rules to govern its administration. This report was adopted by the Board of Directors and became the rules under which the Washington Award has functioned.

Mr. Jackson devoted much time and study to the committee's report, and thereafter to the work of the Washington Award Commission, on which he served for several years, representing at first the Western Society of Engineers and later the American Institute of Electrical Engineers.

In 1918, the two Jacksons and a third brother entered the U.S. military service, necessitating the liquidation of the affairs of the D. C. and Wm. B. Jackson firm and the abandonment for a time of the valuable clientele which had been built up throughout the United States and Canada.

With the rank of major, Jackson's war service was rendered at Camp Merritt, N.J., in operating the utilities of that important point for troop assembly. At the close of the war he was discharged with the rank of lieutenant-colonel, Reserve Corps, U.S.A.

Grounded as he was in the theory and practice of his engineering line, Mr. Jackson also had the true engineering instinct for solving engineering problems not explained in textbooks.

Following are two examples:

In those early days, when the reciprocating engine was the standard prime mover, it was possible to count the rpm of such a loaded engine by noting the dip of the lights which occurred at the end of each stroke of the low-pressure piston, an annoyance bad enough in itself, but which became a serious matter when those internal-speed variations made it impossible to parallel the generating units of a station, thus rendering it necessary to load each unit with a separate group of transmission lines.

In 1897, when Jackson was appointed chief engineer and general superintendent of the Staten Island electric power property, he came into command of a power-generating plant with a number of reciprocating engine-generating units of different types and characteristics in both engine and generator, and because of this diversity it had been found (or deemed) impracticable to parallel the units for electrical output. Through patient tinkering and adjusting of governors, flywheels and electrical connections, these individualistic polyphase units were persuaded to forget their differences and work in harmony on a common bus, with considerable improvement in economy and in the quality of service rendered. This was said to have been the first instance where, under similar conditions, that result had been attained.

Again, when in 1899, Mr. Jackson went with the Colorado Electric Power Company as chief engineer and general superintendent, he found that company furnishing power of polyphase characteristics over a single circuit threading through the mountains from the hydro station to the new and booming gold-mining center at Cripple Creek. As power and light were absolutely essential to carry on those activities, that power was as the breath of life to those customers, who complained bitterly because of frequent service interruptions.

The new superintendent readily ascertained that the service interruptions were due to the failure of defective insulators, but to replace them with the line alive at several tens of thousands of volts was a difficult problem. He, however, devised an original method which made it possible to change the insulators without interrupting service, which method became quite generally used thereafter.

As Mr. Jackson's favorite recreations had been in out-of-door sports and in the solitude of wild country, he longed for the calm of pastoral surroundings after years of stress in engineering and military life, which longing he sought to satisfy on a small farm near Cheshire, Mass. This diversion satisfied him for a short time only. After a year or two he placed a man in charge of his farm and went back to the bustle of engineering, this time with the New York Edison Company as statistician and rate engineer—right-hand man in such matters to John W. Lieb, vice-president and general manager.

This work developed into an important department, which Jackson directed until his steadily mounting blood pressure imposed on him a less strenuous life. He went back to his Cheshire farm to live, but continued in an advisory capacity for his company until shortly before his death, on January 21, 1937, which resulted from heart failure precipitated by a slight attack of pneumonia.

William B. Jackson was the youngest of four children, the oldest being Mrs. Louis E. Reber; the others Col. John Price Jackson, Col. Dugald C. Jackson, and Lieut-Col. William Benjamin Jackson, all of whom are mentioned in Who's Who in America, and all of whom except William are still living. Mr. Jackson was married to Isabel Morris West, of Pittsfield, Mass., September 3, 1903, who survives

[1] Deceased, June 18, 1935. Memorial published in A.S.M.E. Society Records, August, 1936, page 48.

him, as do his four children, Josiah Kennett, John West, Isabel Morrison, and Mary Price.

Mr. Jackson, in his activities in the general affairs of the engineering profession, contributed articles to the engineering press and read papers before engineering societies, some of which are preserved in the Journal of the Western Society of Engineers. He was active in many engineering societies, having been president of the Western Society of Engineers, vice-president and manager of American Institute of Electrical Engineers; vice-president (1915–1917) and manager (1912–1915) of The American Society of Mechanical Engineers, which he joined in 1901; member, American Society of Civil Engineers, National Electric Light Association, and other American and British societies. He was member of various patriotic societies, including the American Legion and the Reserve Officers Association.

Lieut-Col. Jackson's death, stricken as he was while in his mental prime, removes from the stage a gentleman and an outstanding engineer, whose friends throughout the land are counted by legion.—[Memorial prepared for Journal of the Western Society of Engineers by Bion J. Arnold, Mem. A.S.M.E., Albert Reichmann, and Wm. L. Abbott, Mem. A.S.M.E., Chicago, Ill.]

OSCAR FREDRIK JUNGGREN (1865–1935)

The following tribute to Oscar Junggren, published in the *General Electric Review*, November, 1935, was prepared by Ernest L. Robinson (Mem. A.S.M.E.), of the Turbine Engineering Department of the General Electric Company, Schenectady, N.Y.

"Oscar Fredrik Junggren, designer of more than half the world's largest steam turbines, was born in Landskrona, Sweden, on January 10, 1865, and died in Schenectady on September 24, 1935. His father had intended him to be trained in an agricultural school but died when Oscar was young. An uncle decided that he should study at the Institute of Technology in Malmö, from which he was graduated with honors in 1885. His career proved the wisdom of his uncle's choice. Junggren said that he himself had no strong sentiment in the matter at that time.

"He kept his Swedish heritage throughout his life. His character retained many of the best traits of that hospitable, progressive people from whom have come so many mechanical geniuses of first rank. He was extremely frank but tactful, a kindly and warm friend, but reserved in society that did not interest him. Few faces could express so much of sentiment or thought.

"His courage was always that of an enthusiast. He would never go against evidence but he always sought that evidence which would justify what he wanted to do. His was the simple faith that anything one really wishes hard enough to do will really work—that the important thing is the will to achieve.

"Never much of a reader, he was a student of things rather than books. Pictures always attracted him. Technical magazines and publications were of interest to him in proportion as they were illustrated. There were always about his desk bits of metal fashioned into ingenious shapes.

"If Junggren had any real recreation it was probably driving his automobile. But many of his associates doubted whether he had any real recreation other than building bigger and ever bigger steam turbines.

"He was never satisfied with the achievements of last year. Always he planned for higher pressures, for higher temperatures, and for larger machines, continually reaching out in every direction that promised more efficient power generation. Rarely did he seek anything 'cheaper.' His intuition was in close touch with the rapid growth of public utilites, and he truly judged that the industry was keen to accept and pay for every improvement which promised more power or required less fuel.

"To the man in the street, most modern miracles may be completely explained by the simple remark: 'It is done by electricity.' But to the technical man the generation of electric power in large quantities is a different story—a struggle to cope with ever greater forces and difficulties of design, manufacture, and operation in the effort to keep the power networks able to supply the growing demands.

"Approximately two-thirds of the power in the United States is generated by steam. Government records show little change in total coal consumption for the generation of electricity during the past fifteen years in spite of the fact that electric power from fuel has more than doubled. Thus in 1920, 41 million tons of coal produced 27 billion kilowatt-hours while, in 1934, 41 million tons produced 57 billion kilowatt-hours, more than twice as much.

"The American Public is familiar with the growth of the utility business since the war but is not well acquainted with the engineering effort that enabled this increased power output to be developed without an increase in the rate of expenditure of our fuel resources. In the latter accomplishment, Junggren's influence was potent.

"Oscar Junggren was a leader among his associates. Forty years ago as an athletic young man he won the great cross-country 'road race' to Burnt Hills and back by pedalling his bicycle up the Rexford Hill where others walked. No team work was required to win such a victory but to lead in the field of power generation required genius of a higher order. Nobody appreciated better than Junggren the extent of technical knowledge required for turbine design, the necessity for cooperative effort and specialization, and his indebtedness to associates. Asked for advice on plans for a machine closely allied to a turbine in design, he would only shake his head and say: 'Not in my line.'

"Modest in public, he was not given to writing for publication but his letters reflected sound judgment. One who knew him for many years wrote: 'Junggren expressed his ideas in steel and steam rather than in ink and paper.' Many of those best qualified to know agree that to him more than to any other single man is due the credit for the great advances in the power industry during the past twenty years.

"Central-station engineers who came in contact with him as the designer of their most important piece of apparatus knew him as utterly candid in the expression of his opinion as to the ultimate truth, fearless alike of the immediate consequence to his own organization or to that of the operator. Many years experience had taught them that his estimate of ultimate practicability was rarely to be doubted. His judgment as between a minor annoyance and a fundamental limitation of possibilities had to be accepted.

"In the search for more efficient details of design, Junggren was an active and successful leader. He was always an inventor. His patents relating to turbine design, from 1903 and extending throughout his career, numbered 130 and many of them are of basic importance to the art. Junggren had a magic touch that converted his ideas into reality. His interest was magnetic and permeated his associates. He not only had the ability to conceive but the will and the power to accomplish.

"From 1885 to 1889 Junggren was engaged as a draftsman and engineer on steam engines, both for land and marine service, in Christianstad, Sweden. In 1889 he came to the United States and obtained a position with the Edison Electric Company in its office on Wall Street, New York, as a draftsman on maps for the distribution of light in that city, later designing and drafting direct-connected compound- and triple-expansion steam engines. Because of this early connection he was one of the Associate Edison Pioneers.

"In 1890 and 1891 he worked for the Watts-Campbell Company at Newark (N.J.) as a draftsman on Corliss steam engines, and for the Featherstone & Sons Company, Chicago, as a draftsman and engineer on Corliss engines and ice machines.

"In October, 1891, Junggren came to Schenectady and was engaged by the Edison General Electric Company, predecessor of the General Electric Company, as engineer and draftsman on high-speed steam engines for direct connection to generators. He was connected with this work for approximately ten years, during the latter part of which he was engineer in charge.

"When the Curtis steam turbine was actively taken up about 1901 he was made designing engineer on steam turbines and did the designing work on the first large turbine put out in 1903. Later, Junggren was made engineer in charge of the large turbine development of the General Electric Company at a time when the utility industry was entering the period of extremely rapid expansion which continued until after the war. He remained in this capacity until 1922, when he was made consulting engineer in order that his outstanding ability might be devoted entirely to design problems. Many of his greatest achievements were made during the next decade and throughout this period he continued to carry a large share of the responsibility for turbine design. For twenty-five years he was a member of The American Society of Mechanical Engineers.

"Under Junggren's inspiration and direction the Curtis turbine was very early converted from a vertical-shaft to a horizontal-shaft machine and the rapid development in capacity and efficiency which has taken place since this change has demonstrated that this decision was one of vital importance.

"As early as 1916, turbines with conical-shaped steam path and single wheels of increasing pitch diameter were manufactured by the General Electric Company at Junggren's instigation. The subsequent success of this design and its general adoption by other builders show that it is a necessary feature for high efficiency.

"In contrast to the cautious experiments in Europe, Junggren's confidence of success led American utilities as early as 1923 to proceed on a large commercial basis with the adoption of higher steam pressures and the resuperheating cycle. His organization designed the first 600-lb resuperheating turbines in this country and the 60,000-kw machine built in 1924 for that cycle was the largest turbine manufactured up to that time. These units were outstandingly success-

ful, both cycle and mechanical efficiencies having been improved. By 1930 turbines aggregating 1,400,000 kw had been built under Junggren's direction for 600-lb steam pressure, three-quarters of the capacity in this country at that pressure.

"No sooner were the 600-lb turbines in successful operation than Junggren undertook the design of two 1200-lb machines for installation at Boston and Milwaukee. In the face of warnings that theoretical advantages might be lost by excessive leakages, Junggren deliberately set out to make the high-pressure section of the turbine even more efficient than its low-pressure stages. This he accomplished. And these first two installations in 1925 and 1926 were commercially successful from the start. By 1931, 765,000 kw in turbine capacity, designed under Junggren's supervision, was operating in this country at 1200-lb steam pressure. This was 90 per cent of the total capacity operating at these pressures in this country and 80 per cent of that operating in the world.

"The most outstanding development in the turbine field has been the tremendous growth in the capacity of individual turbine units. In this Junggren was a leader from the beginning. There are throughout the world only a hundred turbines of 50,000-kw capacity and larger and of these more than half were built in the Schenectady Works of the General Electric Company under the direct supervision of Oscar Junggren. The others were designed and constructed by effort scattered among eleven other builders throughout Europe and America.

"To keep on year after year extending the design of such large and costly machines beyond the fringes of available experience has required true vision, leadership, and courage of the highest order."

In his "The Autobiography of an Engineer," published in 1931, William Le Roy Emmet (Mem. A.S.M.E.), long consulting engineer for the General Electric Company, in his chapter on the development of the steam turbine, wrote as follows of Mr. Junggren:

"My first activity was to get a design made for a 500-kw unit with turbine and generator. Oscar Junggren had been designing the small engines which we were then building and I knew him and appreciated the fact that he was a strong man with much mechanical skill and ingenuity. He had not thought much of the turbine venture, but I got him on the job and he soon became interested. It is hard for me to assign correctly the credit which is due to Junggren in connection with our turbine development. He has continued to direct our turbine work for years after I gave it up for other activities and has made for himself a national reputation. He was then, and is today, in some respects a more experienced and skillful mechanical designer than I am but he is less of an innovator and much less of an experimenter. I supplied the faith and initiative and dug out all the experimental knowledge upon which our work was based. In our early work for which I felt sole responsibility my ideas were closely followed but it is possible that I might have done better if I had given Junggren more of his own way. He is a man of very strong will and I had many long contests with him. Once after about three years' work he was offered a very attractive position elsewhere. I had told Mr. Rice[1] that I was spending too much of my time contending with him and I could get along better without him. He asked me what he should do about letting him go, and I told him to pay his price and hold him, an action which I have always been glad of. He helped me greatly and I learned much from him and I think that he would say the same of me."

A further tribute to Mr. Junggren comes from R. C. Muir (Mem. A.S.M.E.), now vice-president in charge of engineering at the General Electric Company, Schenectady.

"I first became acquainted with Oscar Junggren in the year 1906 when he was designing engineer of the turbine department of the General Electric Company and I was a student engineer in the turbine test department.

"Normally a test man could not presume to become acquainted with the designing engineer carrying large responsibilities in a large department, but not so with Mr. Junggren. He made every test man feel he was an associate and I soon acquired a respect for him and then a friendship which grew warmer and more intimate during the following 30 years.

"Later I had the good fortune of working under his direction on turbine design and, although I did not fully appreciate them at the time, I have long since come to realize many of his unusual qualifications. As I look back upon his work in the earlier days of turbine development, I feel that he had an intuition or sense of proportions greater than any other engineer with whom I have ever been in contact. I have on many occasions seen him review a newly made layout of a turbine and then draw in, free hand, the contours of the shaft, wheels, diaphragms, and casings which his sense of proportions told him should be correct, and invariably the calculations would indicate he was about right. He had great respect for calculations and the technical aspects of design, and he always backed up his designs by complete calculations and tests, but his power to visualize the complete unit was remarkable.

"He was a delightful man to work under, very liberal with his time and very tolerant of your views. One soon learned to understand his 'um's' and 'ah's,' his 'um' meaning that you had brought to light something which he was actively thinking about, and his 'ah' meaning that he was on the way toward or had arrived at a correct solution. Although in the profession he ranks highest as a mechanical engineer or a designing engineer in the broadest sense of the word, his associates, and particularly the young men working under his direction, will remember his lovable character and the kindly interest he took in their development."

A summary of the qualities which have been brought out in these tributes is offered by Geo. A. Orrok (Mem. A.S.M.E.), consulting engineer, Orrok, Myers & Shoudy, New York.

"Oscar F. Junggren will always represent to me the man who dared to consider the clashing systems of turbine design, picking out from each the points he considered of primary importance and combining them into a new system which had possibilities of economy in construction, economy in the use of heat, rigidity combined with flexibility, and a promise of expansion to the largest sizes without exaggeration in mechanical design. I knew him as an engine designer, as a designer of vertical turbines and as the blender in one machine of many different schools of turbine design to secure an operating machine combining the good points of flexibility, economy, good operation, and successful performance. He made friends easily and kept them. His own generation loved him. The younger generation looked up to him as an inspired leader. While his opportunities were large, his attainments were equal to the problem and his work will live after him."

[1] E. W. Rice, then vice-president of the company, in charge of manufacturing and engineering.

WILLIAM KIELBLOCK (1897–1936)

William Kielblock was born at Neumünster, Germany, on December 31, 1897, a son of Wilhelm and Louise (Schlee) Kielblock. He had been in the United States for a number of years, and attained his citizenship in 1932, but returned to Germany for medical treatment in the fall of 1935 and died in his native city on August 25, 1936.

Mr. Kielblock spent a three-year apprenticeship as machinist and toolmaker with the Deutsche Reichs Eisenbahn and served from 1916 to 1919 in the German army, first as assistant to master of arms and, during the latter part of the time, as master of arms. He then reentered railroad service, working as toolmaker, electrician, and draftsman, and from 1923 to 1925 instructing third-year apprentices in the operation and repair of various types of air and feedwater pumps.

His first position in this country was with Gombar and Rotter, New York, where he spent something over a year in 1925–1926 in the repair and rebuilding of paper-box machinery. Then after a few months at the United Scientific Laboratories, New York, in connection with dies for radio condensers, he became tool and instrument maker for Bernard and Heller, New York. Here he also did some design work and built and tested models for inventors. Leaving that company in the fall of 1928, he spent the remainder of the year on the design and construction of a new type of electric water heater, for the Fireless Electric Boiler Corporation.

With the exception of a few months in 1931, Mr. Kielblock was in the employ of the H. Russell Brands Laboratories, New York, from 1928 to 1933. In 1931 he was assistant foreman at the Fred Goat Company, Brooklyn, making silent turnstiles for subways. At the Brands Laboratories he was in charge of the development of automatic machinery for restaurants.

During these years Mr. Kielblock took evening courses at Cooper Union, securing a B.S. degree in mechanical engineering in 1932. In that year he also became an associate member of the A.S.M.E.; in 1935 he was automatically transferred to the grade of full member.

His work after 1932 was chiefly with Walter Kidde & Co., Inc., in Bloomfield, N.J., for which he designed an automatic press for drawing, trimming, and necking small steel bottles, and various other machinery for use in the manufacture of bottles.

He is survived by his brother, Hugo Kielblock, of Neumünster, Germany.

CHARLES W. G. KING (1862–1935)

Charles W. G. King died at his residence in Chestnut Hill, Pa., on December 21, 1935, of cancer of the larynx. He was born at Northampton, England, on April 5, 1862, son of Charles Dawson and

Sarah King. His education was obtained at a private school and the grammar school of his birthplace.

From 1877 to 1883, Mr. King served as an apprentice in the locomotive department of the Great Northern Railway at Peterborough, England, and remained as a machinist for four years after completing his apprenticeship. He emigrated to the United States in 1887 and became a naturalized citizen ten years later. He was employed by various companies at Philadelphia, Pa., for 48 years. He first worked as a machinist for approximately one year with the Barr Pumping Engine Company. This employment was followed by about six months with William Sellers & Co., as a machinist on electric cranes, and a similar engagement with the machine-tool firm of Bement, Miles & Co.

From 1890 until his death, Mr. King was employed continuously by the United States Metallic Packing Company in the design and manufacture of supplies for locomotives and stationary and marine engines. He was a draftsman until 1899, when he was promoted to chief draftsman and had charge of the design of pneumatic hammers, metallic packing, and pneumatic sanders for locomotives as well as special drilling and grinding machines for their manufacture; in addition, he laid out the shafting and machinery for the company's new plant and supervised the installation. After two years as chief draftsman, Mr. King was appointed superintendent with full charge of the shops and drafting room and held this position for 17 years. From 1918 to 1920, he was president of the company and, at the time of his death, was consulting engineer, a position that he had held for practically 15 years. While with the United States Metallic Packing Company, he secured numerous patents covering metallic packings for locomotives and stationary and marine engines.

Mr. King became a member of the A.S.M.E. in 1906. He was an Odd Fellow and was specially interested in boating, fishing, and mechanical engineering in general. Surviving him are his widow, Carolina (Schladensky) King, whom he married on his thirtieth birthday in 1892; three daughters, Kathryn S., Dorothy M., and Marjorie E. King, all of whom reside in Chestnut Hill, Pa.; and a son, Arthur H. King, of Audubon, N.J.

FRANK LAND (1868–1935)

Frank Land died at his home in Nyack, N.Y., on December 24, 1935. He was born at San Francisco, Calif., on June 25, 1868, son of Charles and Caroline (Powers) Land. He attended private schools in San Francisco and high school in Syracuse, N.Y. He then entered Cornell University and completed the electrical engineering course with the degree of mechanical engineer in 1891. For a short time he was employed by the General Electric Company at Lynn, Mass., and in installation work on the Syracuse trolley system. Subsequently he engaged in automobile parts manufacturing as secretary and treasurer of the I. A. Weston Co., in Syracuse.

From 1903 to 1905 Mr. Land was employed by the Westinghouse Electric & Manufacturing Co. at East Pittsburgh, Pa. Then the Land-Wharton Company was formed at Philadelphia, Pa., with Mr. Land as incorporator and treasurer. He participated in plans for extensions of the Remington Arms Company's plants at Bridgeport, Conn., and Ilion, N.Y. From 1913 to 1915 he was general manager of the Dyneto Electric Company, Syracuse, manufacturing automotive electrical equipment. Following this he engaged in a consulting practice as a mechanical engineer until 1923, when he became associated with The United Electric Light & Power Co., New York, N.Y., in connection with the inventory and valuation of the company's property. Later, with this same company, he was engaged in special research work with the Technical and Engineering Department until his retirement from active work in 1928. He had a special aptitude for the graphical presentation of organizations and routines, and was a mechanical draftsman of the highest order.

Mr. Land had been a member of the A.S.M.E. for 35 years and had belonged to the American Institute of Electrical Engineers since 1891. He was a member of the Cornell Chapter of the Kappa Alpha Society.

Married to Florence Freeman in 1895, he is survived by her and by a daughter, Elizabeth Land.

WILLIAM HARRISON LARKIN, JR. (1871–1936)

William Harrison Larkin, Jr., was born at Worcester, Mass., on April 15, 1871, and died at Arlington, Mass., on July 16, 1936. He attended the Worcester public schools and was graduated from the Classical High School in 1889. He received the degree of Bachelor of Science in 1893 and of Mechanical Engineer in 1903, both from Worcester Polytechnic Institute.

From 1893 through 1901, he was draftsman, machine-shop foreman, master mechanic, and mechanical engineer in the First and Second Lighthouse Districts, U.S. Lighthouse Establishment. In that time he had charge of the equipment, maintenance, and repairs of 187 lighthouses from the Canadian line to Rhode Island. In 1898 he was in charge of laying submarine cables for national defense in the war with Spain. He collaborated with Major William R. Livermore on his "Report on Fog Signals" in 1894.

In 1901 he accepted the position of chief mechanical engineer on the staff of Frank B. Gilbreth, who was at that time carrying on the business of general contractor in Boston, New York, and London.

For several months in 1903, he was the erecting engineer for the concrete machinery at the Howden and Derwent dams of the Derwent Valley Water Project, near Sheffield, England. He was granted a number of patents in connection with his development of concrete machinery.

When the Gilbreth contracting business was discontinued in 1907, he took charge of the boiler shop, foundry, and machine shop of the M. Rumley Co. of La Porte, Ind. From 1910 to 1912, he was executive engineer for the C. W. Hunt Co. of New York.

In 1908, he and his classmate, J. W. Buzzell, of Stone and Webster, were granted the original patent for the conveying of concrete through pipes by pneumatic pressure from point of mixing to the forms. They organized the Pneumatic Concrete Conveyor Company of New York. They carried on experiments at Newark and published the results in an illustrated article appearing November 25, 1909, in the *Engineering News*. Their method of conveying and placing concrete became generally used on such tunnel jobs as the Shandakin Tunnel, Catskill Aqueduct, Holland Tunnels, and many others.

In 1912, he began his long term of service with the United States Rubber Company. He was plant engineer at the Revere Rubber Company, Chelsea, Mass., until 1919; during the World War he built a gas mask division at the Chelsea plant. Also for various periods he supervised engineering projects at the Chicago, Cleveland, and Providence factories. In 1919 he was made power and safety engineer for the Mechanical Goods Division of the company, with headquarters at Passaic, N.J. In 1925 he was made general steam engineer for all plants of the company, with headquarters at New Haven, Conn. During the ten years beginning in 1920, monthly power reports from all plants were instituted, and boiler-room practices standardized. Several new power plants were designed and erected, notably at Detroit, Providence, and Millville.

In 1932, he opened an office in Arlington, Mass., as consulting engineer. In spite of the depression he was servicing more than 30 plants throughout New England and New York by 1936.

He was the author of many articles in various engineering and technical journals, including a number on boiler feedwater, safety, and power published by the U.S. Rubber Co., and contributions on these and other subjects to *Rubber Age*, *Factory*, and *Power*. A paper on "The Supply of Industrial Power," which he presented before the Annual Meeting of the A.S.M.E. in 1925, was published in *Mechanical Engineering* in November of that year.

Mr. Larkin became a member of the A.S.M.E. in 1904 and had served as a member of the Boiler Code Subcommittee on the Care of Steam Boilers and Other Pressure Vessels in Service since its organization in 1922. He had been a member of the Plant Engineer's Club of Boston since its origin in 1915, was a charter member of the Massachusetts Delta Chapter of Sigma Alpha Epsilon, and belonged to the American Society of Safety Engineers and the New England Historic Genealogical Society. Genealogical research and cabinet making were two of his most absorbing interests and he was a pianist and organist of marked ability.

He married at Portland, Me., in 1902, Cordelia G. Fessenden, who with four children, William H., 3rd, Anna C., Mary C. and John E., survives him.—[Memorial prepared by William H. Larkin, 3rd. Mem. A.S.M.E.]

HENRY LOUIS LE CHATELIER (1850–1936)

Henry Louis Le Chatelier, upon whom the A.S.M.E. conferred honorary membership in 1927 for his introduction of new methods of physicochemical analysis, now considered indispensable in all metal-working establishments, died on September 17, 1936. On the occasion of the presentation of honorary membership Professor Le Chatelier referred to the "application of common sense to industry" in America and expressed the wish that the principles employed so successfully in this country might be more universally adopted. His interest in scientific management is brought out by a tribute by Wallace Clark, Mem. A.S.M.E., consulting management engineer, Paris, as follows:

"To his many American friends and admirers Henry Le Chatelier is best known for his translation of Taylor's 'Principles of Scientific Management' and for numerous treatises on those principles, as well as for his long friendship with Taylor.

"It was in 1904, after ten years of determined effort to secure for French metallurgy a Journal comparable with that of the Iron and

Steel Institute, that Le Chatelier succeeded in establishing the *Revue de Métallurgie*. As editor of this *Revue*, he was preparing a series of articles on high-speed steels when he was informed that Taylor had discovered those materials by accident. He then wrote an article stating that chance observations were of little value in science, unless they led to well-planned investigations. This brought a reply from Taylor who explained that his discovery of the steels was not by chance but the result of an intensive study of over two years. He enclosed a pamphlet outlining the methods which he was applying to industrial problems and Le Chatelier was amazed and delighted to find 'examples of the application of the scientific method which might well serve as a model to even professional scientists.'

"Thus started the correspondence and friendship between these two men. In 1909 Le Chatelier translated and published in his *Revue*, Taylor's 'Principles of Scientific Management' which later appeared in book form and met with success which surprised its translator. In 1928 he published 'Le Taylorisme,' which was followed by Polish and Italian editions. These and other books and articles on Taylor methods and on scientific management in general pioneered the way for these new ideas in France, and indeed in Europe.

"But this was only one phase of the career of this French engineer. In the fields of science, research, teaching, and as ingénieur conseil, his record of achievement marks the superman. This has been set forth ably by Ralph E. Oesper, of the University of Cincinnati, and published in the *Journal of Chemical Education* for March, 1931. Another full record appeared in January, 1937, in the Memorial Volume of the *Revue de Métallurgie*. The Le Chatelier writings and discourses listed therein would seem to make a full life for any man. He occupied chairs in the University of Paris, the Collège de France, the Sorbonne, the École des Mines and the École Polytechnique. During his brilliant career as a teacher he covered a wide range of subjects of which these are a few:

Chimie générale (1877);
Chimie industrielle (1896);
Les phénomènes de combustion (1898);
Théorie des equilibres chimiques, les Measures de températures élevées et les Phénomènes de dissociation (1898-1899);
Propriétés des Alliages Métalliques (1899-1900);
Alliages du fer (1900–1901);
Méthodes générales de la Chimie analytique (1901–1902);
Lois générales de la Mécanique Chimique (1903);
La Silice et ses composés (1905–1906);
Quelques applications pratiques des principes fondamentaux de la Chimie (1906–1907);
Propriétés des Métaux et de quelques Alliages (1907).

"When it is considered that actual work and research was always the basis of his teaching and writing, one begins to get an idea of Le Chatelier's prodigious intellect and energy. He made lasting contributions in metallurgy, metallography, measurement of high temperatures, microscopy, ceramics, cements, chemical mechanics, combustion of gases, fuels, and safety in mines, and was an acknowledged authority on steel, glass, cement, and synthetic ammonia.

"In 1922 the leaders of French government, education, and industry came together, with representatives from other nations, at the Sorbonne in Paris, to honor Henry Le Chatelier on his fiftieth anniversary of uninterrupted work and accomplishment in the fields of scientific research, professional work, teaching, and writing. To the high honors and eulogies bestowed upon him at that time, this great and modest engineer made a characteristic reply:

"'All that we are is due only in slight measure to our own labor or to our individual personality. We owe nearly everything to our antecedents—ancestors of flesh and those of the spirit. If one of us adds something to science, art, or morale, it is because a long line of other generations has lived, worked, thought, and suffered before us. Therefore you are honoring today the patient efforts of our predecessors who have created this science.'

In these words I believe is to be found the key to his life and work and to the high esteem in which he was held and will be remembered."

Further biographical data are to be found in the following obituary of Professor Le Chatelier which appeared in *Engineering*, October 16, 1936, page 416:

"The death of Professor Henry Louis Le Chatelier, Doc. ès Sci., which occurred on September 17, at his country house at Miribel-les-Echelles, Isère, France, removes a leading figure from the ranks of physical chemists and metallurgists of international renown. Professor Le Chatelier, who was a former Inspecteur Général des Mines, professor at the Collège de France, Paris, and member of the Académie des Sciences, one of the constituent bodies of the Institut de France, was the eldest son of the late Mr. Louis Le Chatelier, also a former Inspecteur Général des Mines, and was born in Paris on October 8, 1850. He received his education at the Collège Rollin and entered the École Polytechnique in 1869, being the first on the list of the incoming candidates. Graduating in 1871 he passed on to the École des Mines in which he spent three years, and upon leaving, he was appointed on a commission organised by the French Government to investigate the geology of Southern Algeria. In 1877, at the early age of 27, he became professor of chemistry at the École des Mines and, in 1882, was appointed assistant professor at the École Polytechnique. In 1887 he was made professor of industrial chemistry and general metallurgy at the École des Mines, and it was at this stage of his career that he published his well-known work on fuel, entitled 'Le Chauffage Industriel.'

"In 1898, M. le Chatelier was appointed Professor of Mineral Chemistry at the Collège de France, which position he continued to occupy until some years ago, when he was made "professeur honoraire." Few branches of physical metallurgy did not receive his attention and he did great work, more particularly, perhaps, in the realms of pyrometry and metallography. He was responsible for a number of important inventions, among which may be mentioned his metallurgical microscope, thermo-electric and optical pyrometers, and equipment for the investigation of the dilation and electrical resistance of metals and alloys. His name has also been given to a number of metallographic etching reagents which he originated. Of a genial and generous nature, he freely and publicly acknowledged the inspiration and help he had received from eminent contemporaries, among whom he cited Floris Osmond, Sir W. C. Roberts-Austen, Sir Robert Hadfield, and Professors H. M. Howe and A. Sauveur. In recognition of his labours he was elected a member of the Académie des Sciences in 1908, thus filling a vacancy caused by the death of Henri Moissan. He was also Commandeur of the Legion of Honour.

"Professor Le Chatelier was a great teacher and was among the first to broaden the basis of instruction in chemistry. Instead of imparting numbers of isolated facts to his students, he strove to establish the general laws underlying the science of chemistry and indicated how these laws could be applied in particular cases. In this connection he published several interesting works. He was also deeply interested in the application of science to industry and in the laboratory control of industrial products. Furthermore, he was responsible for numerous investigations in the sphere of mining technology, including the determination of the inflammability and other properties of mining gases, the estimation of small quantities of combustible gases and of suspended dust in colliery atmospheres. In May, 1921, his colleagues, numerous friends and old pupils, who had always found him willing to advise and assist them, decided to celebrate his scientific jubilee, 50 years having elapsed since his graduation at the École Polytechnique. An influential international committee was constituted, and a fund opened, in June, 1921, and, by the end of the year, a sum of 170,000 francs had been received. With a portion of this a special gold medal, dedicated to Professor Le Chatelier, was struck, which was presented to him at a gathering held in the chemistry amphitheatre of La Sorbonne, Paris, on January 22, 1922. A surplus of 100,000 francs remained in the fund, and, at the special request of M. Le Chatelier, this was presented to the Académie des Sciences for the creation of a research fund for the promotion of the application of science to industry. The fund bears the name Fondation Henry Le Chatelier.

"In 1910 Professor Le Chatelier was elected an honorary member of the Société des Ingénieurs Civils de France. He was also for many years a member of the Chemical Committee of the Société d'Encouragement pour l'Industrie Nationale, and in December, 1903, founded our esteemed contemporary the *Revue de Métallurgie*, the first number of which appeared in January, 1904. He became a member of the Iron and Steel Institute in 1904, was the recipient of the Bessemer Medal in 1911, and was elected an honorary vice-president of the Institute in 1924. Professor Le Chatelier was made an honorary member of the Institute of Metals in 1912, and was for nearly twenty years honorary corresponding member to the Council of the Institute for France. In 1917 he was awarded the Davy Medal of the Royal Society, of which he became a Foreign Member in May, 1913. He was also for many years honorary member of the Association des Ingénieurs sortis de l'École de Liége (A.I.Lg.), and was the first recipient of the Gustave Trasenster Medal of that body in 1932. The diplome d'ingénieur *honoris causa* of the University of Liége was conferred upon him in 1922, on the occasion of the jubilee celebrations referred to above. He was elected an honorary member of the Institution of Mining and Metallurgy in 1916, of the American Institute of Mining and Metallurgical Engineers in 1905, and of The American Society of Mechanical Engineers in 1927."

ROLLIN CARROLL WELCH LEWIS (1849–1936)

Rollin Carroll Welch Lewis, whose death occurred on March 23, 1936, was born at Harbor Creek, Erie County, Pa., on March 25, 1849, son of Zuriel and Rebecca (Austin) Lewis. He studied for the

medical profession at the Western Medical College, Cleveland, Ohio, at the same time filling an apprenticeship with Thos. Isbister of that city, working on patterns and experimental machinery (1869–1871), and serving as journeyman for the U.S. Screw Co. (1872–1874). After his graduation, however, he turned to industry, rather than medicine, taking a position as master mechanic with the Davies Screw Company, in Dayton, Ohio. In 1878 he went to Cincinnati, Ohio, where for two years he was superintendent of the Wayne Hardware Company.

In 1882 Mr. Lewis secured employment as a toolmaker at the Yale & Towne Manufacturing Co. in Stamford, Conn., and except for two periods he remained with that company until his retirement. He was superintendent of the Springer Torsion Balance Company, Jersey City, N.J., from 1887 to 1892, and from 1898 to 1903 was connected with the Clement Boyer Motor Company, France. He was appointed a juror at the International Exposition in Paris in 1900 and won three medals from the French government for his services to the Exposition.

From 1892 to 1898 Mr. Lewis was foreman of the toolroom for Yale & Towne, and after his return to the company in 1903 served successively as chief draftsman, assistant superintendent of the Bank Lock Department, and superintendent of that department. He retired from active supervision of the department in 1922, but for several years continued to act as its advisor. He held numerous patents covering tools for making combination lock tumblers and one-piece dial, a combination draftsman's table and bench, time locks, and other devices.

Mr. Lewis has been a member of the A.S.M.E. since 1890. He also belonged to the American Association for the Advancement of Science and the National Geographic Society, and was an Odd Fellow, Mason, and Knight Templar. He had received a medal from the Masonic Grand Lodge of Ohio for fifty years of service to the fraternity.

Surviving Mr. Lewis are his widow, Elizabeth S. (Wynkoop) Lewis, whom he married in 1901, and their son, Rollin Carroll Wynkoop Lewis.

RICHARD PAUL LITHGOW (1889–1934)

Richard Paul Lithgow, mechanical engineer for Ford, Bacon & Davis, Inc., died suddenly on September 14, 1934, at Seattle, Wash., while on an assignment for the company there.

Mr. Lithgow was born in New York, N.Y., on December 23, 1889, the son of Richard William and Matilda Lithgow. While taking an evening course in steam engineering at the Y.M.C.A. in New York he worked for the Interborough Rapid Transit Company, New York, as engine-room assistant, and as a watch engineer at the I. Johnson Iron Works, Spuyten Duyvil, New York. Following the completion of the course in 1909 he was employed for four and one-half years in connection with the erection and operation of a pumping plant on the Panama Canal.

After the canal was opened in 1914 Mr. Lithgow took a position as chief engineer for the Arkansas Water Company, a subsidiary of the American Water Works & Electric Co., at Little Rock, Ark. Under his direction the pumping station was rebuilt, much new equipment installed, and the plant brought to a high point of efficiency. In April, 1918, Mr. Lithgow was made traveling engineer for the parent company to look after the operation and condition of the different pumping stations, correct defects in the design of machinery, recommend new equipment where advisable, and in general, see to the efficiency of the stations.

Mr. Lithgow had been associated with Ford, Bacon & Davis, Inc., since 1919. He specialized in valuation work, serving as senior valuation engineer in connection with appraisals for the Puget Sound Power & Light Co., Niagara, Lockport & Ontario Power Co., Lone Star Gas Company, Old Dominion Power Company, Delhart Gas Company, Remington Arms Company, Canada Cement Company, Fox Film Corporation, and other companies.

Mr. Lithgow became a junior member of the A.S.M.E. in 1917 and an associate-member three years later.

He is survived by his widow, Mrs. Helena Lithgow, and by two children, Richard and Eugenia Lithgow.

JOSEPH HUGH McCANDLESS (1882–1936)

Joseph Hugh McCandless was born at Girard, Kan., on November 4, 1882, a son of Rolla Henry and Ruth Jane (Cooper) McCandless. He had a high-school education and later took the complete mechanical engineering course of the International Correspondence Schools, as well as normal, language, and music courses. He served a machinist's apprenticeship, and in 1906–1907 was a student at the California School of Mechanical Arts, at San Luis Obispo.

His first position was that of tracer, blueprinter, and draftsman for the Pacific Steel & Wire Co., Oakland, Calif., following which he did similar work for the White Ornamental Iron Company, also of Oakland. In 1909–1910 he was engaged in work on the Los Angeles Aqueduct, then did mapping for the Western Union Telegraph Company. He was in San Francisco in 1911–1912 working as draftsman for the Joseph Wagner Company, on conveyers, and for the General Petroleum Company on an oil pipe line. Next he made drawings for a new 15-story refinery for the California Hawaiian Sugar Refinery Company, Crockett, Calif., and for power plants and warehouses for the Union Oil Company, San Francisco.

In 1915 Mr. McCandless turned to marine drafting, in the Hull Department at the Mare Island Navy Yard, Vallejo, Calif. He did similar work at the Union Iron Works in San Francisco the following year, then entered the U.S. Air Service as a first lieutenant in the Technical Section. He was stationed at San Diego for a time, later in England and at Paris, where he was executive officer of the Drafting Division.

Following the World War Mr. McCandless worked for a time as draftsman for power plants and on conveyers, in New York, N.Y., then became mechanical engineer and chief draftsman in the Export Department of the Texas Company in that city. He returned to France in 1920 as chief draftsman for James Stewart, Inc., New York, in connection with an oil pipe line from Havre to Paris. He remained in France until 1927, working successively for the Crane Export Corporation on valves and piping; Cie des Entreprises Industrielles, on an invention for refining gasoline; Glacerie de la Chanteriene, Cie St. Gabain, in connection with the construction of a plate glass factory; Cie Nationale des Radiateurs (American Radiator Company); Worthington Pump & Machinery Corp., converting specifications for feedwater heaters to the metric system; and Cie des Constructions Electriques on electric locomotives. He was also assistant superintendent of construction for an automobile highway in the French Alps.

He returned to New York in July, 1927, and was employed by Starrett Brothers there, making composite drawings for the mechanical equipment of buildings. He next did drafting for the Carrier Engineering Corporation of Newark, N.J., and then represented that organization in Paris for a few years. During the latter part of his life he was again located on the West Coast, and at the time of his death on January 18, 1936, was employed in connection with the Bonneville Dam, in Oregon.

Mr. McCandless was unmarried. A brother, Robert M. McCandless, of Hercules, Calif., survives him. He became an associate-member of the A.S.M.E. in 1928.

KENNETH MOLLER (1883–1934)

Kenneth Moller, president since 1930 of the Textile Patent & Process Co., Boston, Mass., died on November 10, 1934. He is survived by his widow, who resides in Milton, Mass.

Mr. Moller was born at Yonkers, N.Y., on December 9, 1883. He received an A.B. degree from Harvard University in 1906 and an S.B. from Massachusetts Institute of Technology the following year He remained at the Institute as assistant in the mechanical engineering laboratory for a year, then went to Providence, R.I., where for about a year and a half he was engaged in the design, construction, and testing of a 250-hp single-cylinder, double-acting, horizontal Diesel engine for the Fuel Oil Engine Company. During the next year and a half he was a salesman for the Commercial Camera Company, of Providence.

In 1913 Mr. Moller became general superintendent and assistant treasurer of the Jenckes Spinning Company of Pawtucket, R.I., a large cotton mill. He had entire charge of the power plant which under his direction was redesigned for the adoption of oil as fuel. This was reported by him as being the first manufacturing plant in New England to adopt oil fuel. During the four and a half years which Mr. Moller spent with this company, approximately two million dollars was expended in plant extensions and improvements, including the electrification of drives.

From 1916 to 1918 Mr. Moller was with the William Whitman Company, Inc., of Boston, in charge of designing, constructing, equipping, and putting into operation a tire fabric mill at Lawrence, Mass. He left this company to enter the service of the U.S. War Department, where he had charge of the purchase, production, and inspection of all clothing for the United States Army.

After the close of the War, Mr. Moller became associated with Lockwood, Green & Co., in direct charge of all its engineering work in the New England district. He made designs and layouts for a large number of cotton mills and was a partner and director of the company when he left it at the beginning of 1925 to become vice-president of the Hunter Manufacturing & Commission Co., New York. After

two years there he accepted the vice-presidency of Jos. Bancroft & Sons, of Wilmington, Del., where he was located until 1930.

Mr. Moller had been a member of the A.S.M.E. since 1921.

AUGUSTUS F. NAGLE (1841–1920)

It was not until 1931 that the A.S.M.E. learned of the death in 1920 of Augustus F. Nagle. Information for a complete memorial notice could not be secured, therefore a brief statement was published at that time. There has since come to light an autobiography written by Mr. Nagle in 1895, which is reproduced here for permanent record.

"I was born in Germany on August 4, 1841, and came to this country in 1851.

"I was educated at the Fredonia Academy, New York, for a civil engineer, but followed the German custom of first learning a trade and therefore served a three years' apprenticeship in a machine shop.

"In 1863 I entered the regular service of the U.S. Navy, but resigned at the close of the war to engage in private work. In 1867 I went to Providence, R.I., and for some years was connected as draftsman and mechanical engineer with the construction of the Providence Water Works. I designed the high-service pumping engine for these works. This is a geared engine, of the vertical compound type, with cranks set at 180 deg. It was the first 180-deg compound engine built in this country, and the first compound engine applied to direct-service pumping works. It gave a duty of 84,673,245 ft-lb per 100 lb of coal, with an evaporation of less than 8 lb of water per pound of coal. This was the highest duty attained at that time by a direct-service pumping engine.

"In 1882 I went West and introduced the Grinnell automatic sprinkler, previous to which time no automatic sprinkler had been in successful use in the West. For nine years I was connected with the sprinkler business, after which I was engaged in purely professional engineering work.

"In 1894 I became connected with the Metropolitan West Side Elevated Railroad Company, Chicago, as consulting engineer, with special work relating to its large power station. I have taken pride in the simplicity, effectiveness, and economy of this work. In this work I am engaged at the time of writing this sketch.

"If I may sit in criticism of my own work, it is that my taste and judgment have always been more that of a mathematician than of a mechanic. At the early age of 15 I had already taken the complete mathematical course at the Academy, including the Calculus, and all through life my inventions and work have been designed on fine, or what is called theoretical, lines, and only by the greatest care and effort have I been able to bring it to practical everyday uses.

"Although educated in this country and constantly associated with mechanical work, the cast of my mind is of the characteristic German metaphysical type, and I love to study abstract problems.

"In character, I am a purist, and hence lose some of the personal charms attached to men given more to good-fellowship.

"I am of a religious nature, but hold no longer to the dogmas of any church.

"I have been married twice, and have living a wife and a little son 9 years old. I became a member of the A.S.M.E. in 1880."

HENRY SHOEMAKER NEWLIN (1887–1935)

Henry Shoemaker Newlin, who became an associate-member of the A.S.M.E. in 1921, was born at McKeesport, Pa., on October 8, 1887, son of Theodore F. and Harriet (Shoemaker) Newlin.

Following his graduation from the McKeesport High School he was employed for about two years in various capacities in connection with the construction of new skelp mills at the McKeesport works of the National Tube Company. He then attended technical school for a year, after which he returned to the National Tube Company, where he worked on electric traveling crane construction and maintenance until the fall of 1909.

During the next three years Mr. Newlin was construction engineer for the Pennsylvania Steel Company at Steelton, Pa. At the beginning of 1913 he became chief engineer at the Sharon (Pa.) plant of the American Steel Foundries. He remained there until April, 1918, during which period he increased the plant capacity from 1000 to 5500 tons per month, necessitating the construction of four new open-hearth furnaces, together with buildings and equipment necessary to handle the increased tonnage. He also had charge of plant maintenance.

Following the termination of this work Mr. Newlin was engaged for a time in private experimental and business matters. In August, 1919, he became production manager of the Sharon Pressed Steel Company. He was with this company until early 1921, his work including the construction of a new plant at Sharon. He next took the position of superintendent in charge of design and construction of a large scrap yard for the Rotter Speer Company at Sharon. He left there in 1923 to engage in research and experimental work for the A. O. Smith Corp., Milwaukee, Wis., and in 1925–1926 carried on research work for the Wm. Millspaugh Paper Mills at Sandusky, Ohio.

From May 1, 1926, until July, 1928, Mr. Newlin was engaged in special experimental and development work in the Jersey City plant of The M. W. Kellogg Company, and after that time and until his death on September 16, 1935, was sales engineer for the company.

He is survived by his widow and six children.

THEODORE OLUF NICKELSEN (1881–1934)

Theodore Oluf Nickelsen died at the Singapore General Hospital on August 26, 1934. He had been on his way to Manila from India, where he had spent several months on business, but severe illness necessitated his removal to the hospital when he reached Singapore.

Mr. Nickelsen, who had been engaged in engineering work in Hawaii and the Philippines for nearly thirty-five years, was born in Liverpool, England, on September 23, 1881, son of Jens Oluf and Amy Mary N. (Lemmer) Nickelsen. At the age of seventeen he entered the employ of the Mirrless Watson Engineering Company in Glasgow, Scotland, where he remained for about two years. He went to Kihei, Maui, T.H., early in 1902 in connection with the development of a deep well pumping plant and during the next ten years served a number of companies in erecting and operating capacities. As chief operating engineer of a Honolulu cane sugar mill and refinery for the last five years of this period he made improvements resulting in the reduction of fuel consumption, introduced the first steel mill housing designed and manufactured by the Honolulu Iron Works, and modernized equipment.

Mr. Nickelsen went to the Philippine Islands in the fall of 1913 to assist in the erection of new equipment at the Calamba Sugar Estate, Laguna. Subsequently he was managing engineer for the Philippine Sugar Estate Development Company at Calamba until 1916. He then established an office as contracting engineer in Iloilo, Panay. He was engaged to dismantle a 250-ton sugar factory at Tigbauan, Panay, transport it to Ilog on the Island of Negros, and re-erect it there, a task which he performed in two and one-half months. He later erected several mills and designed new equipment for other companies.

After traveling in Japan, China, and the United States in the winter of 1918–1919 Mr. Nickelsen returned to Honolulu for a short time, being connected with the Honolulu Iron Works. He then became chief engineer under the contractor for the erection of a sugar mill at La Carlota, P.I., upon the completion of which he was retained as chief consulting and operating engineer. After three years there, during which time he installed additional equipment, he spent about six months as chief erecting engineer for the Del Carmen Sugar Central, Luzon, and then was engaged for two seasons in the remodeling of the factory of the Mindoro Sugar Company, bringing the equipment up to a higher efficiency and increasing the capacity of the plant.

During the last ten years of his life Mr. Nickelsen was connected successively with the Kabankalan Sugar Company, Inc., and Ma-ao Sugar Central Co., Inc., Occidental Negros, the Asturias Sugar Central, Inc., San Juan, Dumalag, Capiz, and Earnshaws Docks and Honolulu Iron Works, Port Area, Manila. He became a member of the A.S.M.E. in 1927 and was a past-president of the Philippine Society of Sugar Technologists and a member of the Philippine Sugar Association. He had contributed articles on the sugar industry to periodicals in Hawaii and the Philippines. He is survived by his widow, Meta (Petersen) Nickelsen.

LARS GUSTAF NILSON (1862–1935)

Lars Gustaf Nilson, a member of the A.S.M.E. since 1916, died at his home in Hoboken, N.J., on November 29, 1935. He was a native of Sweden, having been born at Skattkärr on September 8, 1862, the son of Nils and Caroline (Larson) Peterson. He attended public school in Sweden and after coming to the United States pursued his studies in evening school and in private.

The Edison Pioneers, of which Mr. Nilson was an associate member, has issued a memorial containing the following record of his professional experience:

"He became associated with the Edison interests on October 1, 1886, when he was employed by A. S. Vance, superintendent of the Des Moines Edison Light Company, Des Moines, Iowa, as a meterman. He was soon given assistants so that he could devote his time to other work, such as installing motors, locating trouble, repairing engines and dynamos, etc., and was successful in rewinding burnt out

"H" armatures. In 1888 Mr. Vance resigned and Mr. Nilson was appointed superintendent. In connection with the building of a new station Mr. Nilson assisted in planning the power house, system of distribution, etc.

"From August 1, 1890, to February 1, 1893, he was superintendent of the Sioux City Electrical Supply Company, Sioux City, Iowa, during which time he installed a number of isolated plants, using Edison machines. He planned and had charge of the lighting of the Sioux City Corn Palaces and Carnivals in 1890 and 1891, one of the earliest examples of ornamental electric lighting. In 1893 he became superintendent of the Sioux City Brass Works, where he designed and built direct-current dynamos and motors up to 50 hp.

"From February 1, 1898, to January 1, 1900, he was chief engineer of the Patton Motor Company and S. M. Fischer, both of Chicago, where he did gas-electric pioneer work; and from January 1, 1900, to September 1, 1904, he was chief engineer of the Fischer Motor Vehicle Company of Hoboken, N.J., working on gas-electric trucks and omnibuses. From that date until February 1, 1909, he acted as chief engineer of the Strang Gas-Electric Car Company of New York, manufacturers of gas-electric rail cars.

"In 1909 he became chief engineer of the Nilson-Miller Company, Hoboken, N.J., working on special machinery and tools, gear cutting and general machine work, in which capacity he served until January 1, 1927, when he became a consulting engineer with his headquarters at 1219 Garden Street, Hoboken, N.J."

Mr. Nilson held a number of patents on electrical and mechanical appliances, automatic telephone switches, gas-electric automobiles, railway cars, internal-combustion engines, the Nilson "Kritiscope" "Volute" car brake, a hydraulic dynamometer, gas-engine valves, piston rings, and other inventions and developments.

Mr. Nilson had been married twice, first to Huldah S. Anderson, of Des Moines, in 1907, and in 1927 to Florence M. Gahagan, of Jersey City, N.J. He became a member of the Society of Automotive Engineers in 1910 and served as treasurer of its Metropolitan section in 1919–1920. He belonged also to the American Society for Steel Treating (now the American Society of Metals), the New York Electrical Society and New York Railroad Club, the American Society of Swedish Engineers and the Swedish Engineers Society of Chicago, and was a member of the Masonic fraternity.

WERNER NYGREN (1868–1936)

Werner Nygren, president of Werner Nygren, Inc., consulting engineers in the field of steam power, heating and ventilating, New York, N.Y., died of heart failure in Freeport, L.I., on August 29, 1936. Four children, two sons and two daughters, survive him. His wife, Wilhelmina C. (Magnusson) whom he married in 1891, died in 1935. One of his sons, Alfred M. Nygren, is a member of the firm which he founded in 1927.

Mr. Nygren was born at Billingsfors, Sweden, on July 24, 1868, son of Andrew and Sara Elizabeth (Unge) Nygren. He attended public and high school in Billingsfors, then went to Vänersborg for his technical education. He was graduated from the mechanical engineering course in 1888 and after working for a short time as a draftsman in Sweden, came to the United States, where he was first employed by B. Davidson, architect, Lynn, Mass.

In 1890 Mr. Nygren began several years' employment with the Thomson-Houston Electric Company, at Lynn, as draftsman and engineer on steam and mill work. While with the company he prepared the design for the steam-power and heating equipment of its wrought-iron casting foundry at South Boston. In 1893 he was connected for a short time with Fred Tudor, consulting engineer, Boston, preparing the design for the heating and ventilating of the Boston City Hospital Surgical Building and other buildings, then took a position in the office of Edmond Wheelwright, city architect of Boston, as draftsman, working on heating and ventilating designs for the public schools.

Mr. Nygren took up work in New York in November, 1895, as assistant to Alfred R. Wolff (Mem. A.S.M.E.). This association continued until the death of Mr. Wolff in January, 1909, then the firm of Nygren, Tenney & Ohmes was formed to continue his practice. Mr. Nygren was managing partner of the firm until 1916, and from that time until the formation of Werner Nygren, Inc., engaged in private practice in the field of steam power, heating, and ventilation. During his many years in this field in New York, he was in charge of the installation of equipment for many of the important buildings in the city.

Mr. Nygren had been a member of the A.S.M.E. since 1914. He was past-president of the New York Association of Consulting Engineers and the Heating Board of Trade of New York, a member of the American Society of Swedish Engineers, New York Building Congress, Metropolitan Museum of Art, Engineers' Club, New York, and belonged to the Elks and Masonic orders. He served as chairman of the Divisional Committee on Heating and Ventilation of the Advisory Commission of the Council of National Defense in 1918. Gardening and fishing were his chief diversions. He became a citizen of the United States in 1897.

FRANK JOHN O'LEARY (1868–1934)

Frank John O'Leary, a member of the A.S.M.E. since 1911, died of angina pectoris in Detroit, Mich., on April 1, 1934. He is survived by his widow, Marium (Fisher) O'Leary, whom he married in 1897, and by three daughters, Elizabeth O'Leary, Ellen McCrum, and Frances Faucher, all of Detroit; also by a sister, Anna M. O'Leary, of Detroit, and a brother, of Redford, Mich.

Mr. O'Leary was born at Canandaigua, Ontario County, New York, on February 20, 1868, son of Humphrey and Ellen (Cooney) O'Leary. He attended public school in New York and Michigan and took private instruction in drafting. At the age of twenty he entered the employ of the Detroit Lubricator Company, with which he remained until the spring of 1894. During the next four years he was employed by the Penberthy Injector Company, Detroit, which recalled him in 1907 to take charge of experimental and mechanical work and retained his services as mechanical expert for a period of five years. Between the years 1898 and 1907 he was connected with the Lee Injector Company, Detroit, and the William Powell Company, Cincinnati, Ohio, and from 1913 to 1917 was manager of the Special Injectors Company, Detroit. All of these firms were engaged in making brass steam supplies, and Mr. O'Leary improved the designs and manufacturing equipment and methods. During part of 1912–1913 he was on the time-study staff of the Packard Motor Car Company.

In 1914, Mr. O'Leary also became associated with the appraisal work being carried on by Dean Mortimer E. Cooley (Mem. A.S.M.E.), of the University of Michigan. In 1914–1915 he made an appraisal of the Detroit United Railways, and in 1915–1917, appraisals of the Public Service Railway Company and Public Service Electric Company, Newark, N.J. During the next few years he was assistant engineer to Harvey E. Mole (Mem. A.S.M.E.), of New York, N.Y., and in 1921–1923 made an appraisal of the Public Service Gas Company, Newark, for Dean Cooley. In all of his work for the Cooley organization he had complete charge of the field inventory and pricing of all shop equipment and allied items.

At various intervals between November, 1923, and June, 1927, he was employed as valuation engineer by the National Appraisal Company of Boston, Mass. He then became connected with the firm of Cornwall & Stevens, insurance brokers, of New York, for which he was traveling engineer until the time of his death.

JAMES BLAINE PATTERSON (1890–1936)

James Blaine Patterson, Midwest representative of the A.S.M.E., died on October 16, 1936, in the Presbyterian Hospital in Chicago, following an operation for acute appendicitis.

Mr. Patterson had been advertising representative of the Society in the Chicago area since 1925. Following the death of R. R. Leonard, Midwest representative of the Society, in December, 1935, Mr. Patterson had also taken over his duties in the Chicago office. His wide acquaintanceship and experience in the advertising field, where his work had been since 1916, coupled with his keen devotion to the interests of the Society, made him a much appreciated and valuable member of the staff.

Mr. Patterson was born at Camden, N.J., on October 30, 1890, son of George Washington and Laura (Grapewine) Patterson. He attended high school at Yonkers, N.Y., and studied at the College of the City of New York for one and a half years. He then entered the Carnegie Institute of Technology, which conferred a B.S. degree in mechanical engineering upon him in 1914. During the next year he was graduate assistant in the engineering laboratories there.

For a time in 1915–1916 Mr. Patterson was connected with the Public Service Company of New Jersey, engaged in temporary work in Trenton and Camden. He then became associated with the P. H. & F. M. Roots Co., Connersville, Ind. (now the Roots-Connersville Blower Corporation), and after spending about six months in the shops became their Chicago district manager and also advertising manager. It was in this position that he built up his knowledge of both engineering and sales work, as well as a large circle of acquaintances in the industrial field. For a year prior to his connection with the A.S.M.E. he was engaged in sales engineering as a member of the Hopp-Patterson Company, Chicago.

Mr. Patterson married Miss Bernice Cummings of Chicago, in 1919, and is survived by her and by three children, John, Marjorie, and Robert. A sister, Mrs. D. L. Evans, of Grand Rapids, Mich., also survives him.

A resident of Park Ridge, Ill., since 1926, Mr. Patterson was an active political and civic leader there. He was a charter member and past-president of the Better Government Club of Park Ridge. In the fall of 1932 he formed the Pickwick Publishing Company, which published the Park Ridge *Advocate*, a weekly newspaper.

Mr. Patterson was also past-president of the Engineering Advertisers Association, Chicago. He became a member of the A.S.M.E. early in 1936. He joined the Phi Sigma Kappa fraternity while at the College of the City of New York, and he was a Mason.

HUGH PATTISON (1872–1936)

Hugh Pattison, who died on August 20, 1936, was born on August 3, 1872, near Cambridge, Md., the son of John Richard Pattison and Emily (DeValin) Pattison and a descendant from three of the earliest Colonial Maryland families—all early settlers in Dorchester County, namely, the Pattisons, the LeCompts, and the Skinners. Of these, Thomas Pattison first settled in Calvert County, but moved to Dorchester County in 1671. He was clerk of the Court of Dorchester County and, in 1688, was appointed his Lordship's attorney for that county. The home farm, about four miles from Cambridge, where Hugh Pattison and his brothers and sisters were born, has been left as part of his estate.

Mr. Pattison attended the McDonogh School for boys near Baltimore, Md., from June, 1883, to December, 1889, with a record as one of the outstanding students in ability and character, very much interested in machine-shop work, and excelling as a draftsman. In January, 1890, he entered Johns Hopkins University, at Baltimore, for the course in electrical engineering, which he completed in June, 1892, receiving a Certificate of Proficiency in Applied Electricity (P. A. E.) such as all graduates in the electrical engineering course received at Johns Hopkins in those days. In 1927 he was awarded a diploma as Bachelor of Science *Extraordinem*, by the same institution.

During his college vacations, Mr. Pattison worked for the Baxter Motor Company, of Baltimore, as armature winder and as draftsman. After his graduation, he obtained employment in the Equipment Department, at the United States Navy Yard, at Norkfolk, Va., and remained there several months in charge of the installation of wiring on three cruisers—some of the earliest, if not the earliest, electrical installation work on American war vessels. Then, with Sprague, Duncan & Hutchinson, Ltd., consulting engineers, he was assistant engineer in the Baltimore office of the firm, engaged on the work of preparing plans and specifications and supervising electrical installations in office buildings and on the vessels of the Merchants & Miners Transportation Co. In 1894, he was placed in charge of the construction of the electric plant and all wiring work in the Congressional Library, at Washington, D.C., and after completing this work, stayed on in charge of operation of the plant until 1898.

Mr. Pattison then accepted a position with the Sprague Electric Company as secretary to the vice-president and technical director, in which capacity, for some time, he followed office work in connection with the first multiple-unit electric-train equipment on the South Side Elevated Railway in Chicago, Ill., the Brooklyn (N.Y.) Elevated Railway, the elevator equipments on the Central London (underground) Railway, and, later, in charge of installing and testing multiple-unit railway equipments in Brooklyn and in Boston, Mass.

In 1901, he left the Sprague Electric Company to accept a position as resident engineer (representing the consulting engineer) in charge of the design and erection of a new power plant for the Pennsylvania Steel Company (later, Bethlehem Steel Company), at Steelton, Pa., including substations, high-tension transmission lines, and the lighting and motor equipment in the new shops.

After the completion of the work at Steelton in 1903, Mr. Pattison went with Westinghouse, Church, Kerr & Co., New York, N.Y., as engineer, and with that company, was engaged on the foundation work of the Long Island City Power House and later on the design and installation of electric car equipment and the erection of car inspection shops for the Long Island Railroad Company. After the completion of this work, he was made assistant engineer of electric traction with the Pennsylvania Tunnel & Terminal Railroad Co. and the Long Island Railroad Company, and was placed in charge of the electrification of the West Jersey & Seashore R.R., from Camden to Atlantic City, N.J. Later, he had charge of the construction and operation of an experimental single-phase, alternating-current, electric railway on Long Island for the purpose of obtaining information about alternating-current equipment to help in deciding the question as to whether direct current or alternating current should be used for the Pennsylvania Railroad tunnel electrification. During 1907, he served as field engineer in charge of a series of tests on the West Jersey & Seashore R.R. made by the Pennsylvania Railroad Company to determine whether there were any features in the known designs of electric locomotives which might introduce strains on the track in excess of those exerted by steam locomotives, and to obtain information which eventually would lead to a mechanically satisfactory design of electric locomotive. Side pressures on rails were measured by the impressions made on steel plates by hardened steel balls, with ties of special construction having roller bearings under the rail supports. High-speed tests were made with different types of steam and electric locomotives.

Mr. Pattison was next appointed superintendent of construction (under the chief engineer of electric traction) in charge of the construction work in connection with the electrification of the Pennsylvania Railroad in the New York area, including the foundations and the erection of steel transmission-line towers across the New Jersey meadows from Manhattan Transfer through the New York Terminal to Sunnyside Yard, making and erecting reinforced-concrete telegraph and telephone line, all equipment in substations, various electric circuits in the tunnels, station, etc.

After the completion of the Pennsylvania Railroad Terminal in 1911, Mr. Pattison was appointed electrical engineer for the Chicago Association of Commerce Committee of Investigation on Smoke Abatement and Electrification of Railway Terminals, in Chicago. He remained in this position in exhaustive studies until the completion of the work in 1915. Then, from 1915 to 1919, he was electrical engineer, superintendent of inspection (during the World War), and assistant to the general manager of the Eddystone Rifle Plant, at Eddystone, Pa.

For the next two years Mr. Pattison was engaged in making studies and analyses of existing steam railroad electrification, including the Chicago, Milwaukee & St. Paul Ry., the Norfolk and Western, the New York, New Haven and Hartford, and the Pennsylvania railroads, and making reports on comparative operating and maintenance costs for the Westinghouse Electric & Manufacturing Co.

During 1921 and 1922, Mr. Pattison served as electrical engineer of the Electrification Commission for the Illinois Central Railroad, Chicago, to decide upon plans for and the system of electrification to be used. He was a delegate to the International Railway Congress which met in Rome, Italy, in April, 1922. After the completion of the work of the Illinois Central Railroad Commission, Mr. Pattison spent several months preparing a proposal for the electrification of the Virginian Railway, and, then, from 1923 to 1933, he was engineer of electric traction of the Virginian Railway Company, having general engineering supervision over the electrification work between Mullens, W.Va., and Roanoke, Va., covering power plant, transmission and overhead lines, substations, and locomotives during the construction period, and, subsequently, over the electrical operation.

In 1933, Mr. Pattison accepted the position as engineer of car equipment with the Third Avenue Railway Company, New York, at the time when this company was embarking upon a new, extensive program of rehabilitation of cars and equipment. He had charge of the drafting-room work and was responsible for all revisions in car and equipment designs and for new designs. Owing to his wide experience, Mr. Pattison was also consulted on the technical side of almost all engineering projects involved in the Mechanical and Building Departments of the Third Avenue Railway Company. His last important work was the supervision of the designs of entirely new light-weight intermediate side-door cars of aluminum alloy and steel alloy, respectively, which may prove to be the ultimate standards to be adopted. He was busily engaged in this work until he went to the New York Hospital, less than a week before his death, for an operation from which he did not recover.

As a man and as an engineer, Mr. Pattison always commanded the respect and admiration of his associates and of men under him. He was eminently fair and just in all his dealings, always thoughtful and considerate, and always ready with encouragement or help. He became deeply interested in any task or engineering problem which his duties imposed, went about his work in a quiet and unassuming manner, doing what he believed was right, and his example was always a source of inspiration to his coworkers. He had a remarkable memory for incidents and details, the gift of expressing his ideas clearly, a keen sense of humor, and, as a conversationalist, was always entertaining.

One of Mr. Pattison's associates in the Third Avenue Railway Company has written about him as follows:

"Hugh Pattison accepted the position of engineer of equipment in the Mechanical Department of the Third Avenue Railway Company in June, 1933, just when this particular department greatly needed attention. He brought to the position technical ability of the highest order and a rich background of experience, but, above all, he brought a fine influence in the spirit in which he did his work and in the steadily splendid example which he set. His loss is one that you and I and the world at large can ill afford. There are entirely too few men like Hugh Pattison."

Dr. Cary T. Hutchinson, of the early firm of Sprague, Duncan &

Hutchinson, Ltd., with whom, later, Mr. Pattison was associated at different times, has written as follows:

"Long association with Hugh Pattison beginning in 1893 and continuing well through his life saw the development of a man of the highest integrity, both personal and intellectual. He never deviated from what he thought was right, no matter how great the pressure upon him might be. His loyalty to friends was unfailing and his modesty was extreme. His engineering knowledge and ability were of the highest order. It will be hard to find his equal. His friends will long remember him."

Mr. Pattison was never married. He is survived by two brothers, Judge John R. Pattison, of Cambridge, Md., for many years a judge of the Court of Appeals of Maryland, and James B. Pattison, of Philadelphia, Pa., and by several nieces. Former Governor Robert E. Pattison of Pennsylvania was a first cousin.

He became a member of the A.S.M.E. in 1914 and was also a member of the American Institute of Electrical Engineers and the American Society of Civil Engineers.—[Memoir prepared by Alex McIver, New York, N.Y., for the Transactions of the American Society of Civil Engineers.]

WALTER GRANT PENFIELD (1872–1934)

Walter Grant Penfield was born on November 11, 1872, at East Berlin, Conn., son of Walter Edwin Penfield, a machinist, and Joanna (Weed) Penfield. He studied civil engineering at Cornell University from October to December, 1890, and attended Williston Academy, at East Hampton, Mass., from 1892 to 1894. He then entered the Sheffield Scientific School at Yale University. He left there in his senior year to take a position as draftsman in the bridge department of the New York, New Haven & Hartford R.R. A Ph.B. degree was conferred upon him by the University in 1905, with enrollment in the Class of 1896 S.

From May, 1898, when he enlisted in the 1st Connecticut Heavy Artillery, U.S. Volunteers, until after the close of the World War, the greater part of his time was spent in army service. He was promoted to first sergeant and second lieutenant before being mustered out in October, 1898. After working again during the winter as draftsman in the bridge department of the New York, New Haven & Hartford R.R. at New Haven, he was commissioned second lieutenant of infantry in April, 1899, assigned to the 1st Infantry. He was stationed for short periods at Columbus Barracks, Ohio, and Governor's Island, N.Y., and was at Pinar del Rio, Cuba, from May, 1899, to August, 1900. He served with his regiment at Manila, P.I., in 1900–1902, and also as engineer officer with the 6th Separate Brigade. He was promoted to first lieutenant in February, 1901, and to adjutant, 3d Battalion, in January, 1902. In June of that year he was detailed to the Ordnance Department and was stationed at the Sandy Hook Proving Ground until August, 1903, at the Rock Island (Ill.) Arsenal in 1903–1904, and at the Watertown (Mass.) Arsenal in 1904–1906. He was in command of Company C, 14th U.S. Infantry, at the Vancouver (Wash.) Barracks in 1906–1907, and on duty at the U.S. Military Prison, Alcatraz Island, San Francisco Bay, during April–July, 1907. He was then detailed a captain in the Ordnance Department, and spent the next seven years at the Springfield (Mass.) Armory, being promoted to the rank of captain, infantry, in 1908, and detailed major, Ordnance Department, in October, 1910. In 1914 he was appointed president of a board of officers to select a new type of machine gun for the United States Army and was transferred to the Frankford (Pa.) Arsenal, where he was in charge of the manufacture of artillery ammunition until his resignation in 1915.

During the World War he was connected first with the Remington Arms Union Metallic Cartridge Company, at Bridgeport, Conn., and Wilmington, Del., and later with the Trexler Company of America, Philadelphia, of which he was vice-president. He also served as lieutenant-colonel of the 2d Regiment of the Connecticut Home Guard. After the war he was associated with the Davis-Warner Arms Corporation at Norwich, Conn., and the Steiger Realty Corporation, Hartford, Conn. He served as president of the Sturdi-Truck Manufacturing Company, a corporation of Delaware, with factory at Greene, N.Y., in 1925–1926, and continued as a director of the company until 1933. Since 1931 he had been construction engineer for the U.S. Treasury Department. He was in charge of post-office construction at Exeter, N.H., in 1932, at Cincinnati in 1933, and at Detroit at the time of his death, which occurred at Hartford, Conn., on January 3, 1934.

Major Penfield has been a member of the A.S.M.E. since 1919 and belonged to the Yale Engineering Association and to the Yale Club, New York. He is survived by his widow, Adeline Mary (Burnham) Penfield, whom he married in 1896, a nephew, Jesse R. Penfield, and a sister, Susan (Penfield) Hodgson.

GUY S. POLING (1888–1935)

Guy S. Poling, a partner in the firm of Poling and Avery, New York, N.Y., was killed on March 13, 1935, in a fall while inspecting the Bushwick Theater, in Brooklyn, N.Y. The firm of Poling and Avery was formed in 1931 and handled a wide variety of valuation projects for industrial public utilities and municipalities.

Mr. Poling was born in Brooklyn on August 16, 1888, the son of Frank L. and Lina (Nostrand) Poling. He was educated at the Boys' High School, Brooklyn, and at Cooper Union, New York. He began work at the age of sixteen and during the next six years was engaged in engineering, surveying, and construction work in New York as instrument man, inspector, and chief of party.

In 1911 and 1912 Mr. Poling was assistant superintendent of construction of the Los Angeles Aqueduct, Narka Division, and assistant engineer for the Los Angeles Mesa Development Company. Then for a year and a half he was chief engineer of the Elsinore Development Company and City Engineer of Elsinore, Calif. He laid out and developed some 8000 acres for ranches and townsites, purchasing and erecting machinery for pumping plants, water supply, and irrigation systems; and completed the construction of a sewage-disposal system for Elsinore, installed a new pumping plant, and laid pipe lines for a water-supply system.

From August, 1914, until the United States entered the World War in April, 1917, Mr. Poling was chief engineer for the Pacific Oil & Cotton Co., Calexico, Calif. He designed, purchased materials for, and constructed the buildings for 24 cotton gins, 2 cottonseed oil mills, storage and warehouse buildings, offices and residences. He also designed the layout, purchased and erected all the machinery for the plant, and maintained a complete machine shop for mechanical and auto truck repairs. When construction was completed he had charge of operation and maintenance of the plant.

For two years, beginning in May, 1917, he served as a captain of the Field Artillery, U.S.A. He returned to private life as a valuation engineer, practicing in New York under his own name until the formation of Poling and Avery.

Mr. Poling became a member of the A.S.M.E. in 1926 and also belonged to the American Society of Civil Engineers. He is survived by his widow, Jessie W. Poling, and their daughter, Barbara L. Poling.

BERNARD EUGENE PRATHER (1885–1935)

Bernard Eugene Prather, whose death occurred on January 14, 1935, was born at Waverly, Ala., on March 13, 1885.

His early education was obtained in the rural schools of Alabama. He attended the Alabama Polytechnic Institute and was graduated as a Mechanical Engineer in 1907. After graduation he served his student apprenticeship with the Allis-Chalmers Manufacturing Company, Milwaukee, Wis., and after completion of this course in 1909 entered the Service and Erection Department as an erecting engineer. In 1918 he was appointed assistant general superintendent of the Service and Erection Department of this company, which position he held until his death.

Mr. Prather became a member of the A.S.M.E. in 1922. He was a member of the various Masonic bodies, Engineers Society of Milwaukee, and Westmoor Country Club.

Mr. Prather is survived by two daughters, Marian and Edith, his brother Clyde, and a sister, Mrs. George Moxham, all of Auburn, Ala.

JOHN MACKLIN RATHMELL (1880–1935)

John Macklin Rathmell, a descendant of the pioneer settlers in Pennsylvania, died at Aurora, Ill., of angina pectoris on December 13, 1935. He was born at Williamsport, Pa., on August 11, 1880, son of Joseph Guilky and Alice (Van Dyke) Rathmell. For two years he attended Susquehanna University, Sellinsgrove, Pa., and supplemented this by International Correspondence Schools courses in mechanical and electrical engineering.

In 1903, he secured a position as draftsman with the Valley Iron Works, Williamsport, and, when he left in 1906, had been promoted to chief draftsman. Practically all of the next year was spent as a draftsman and layout man with the General Electric Company at its Schenectady, N.Y., plant. From 1907 to 1910, Mr. Rathmell had a number of positions with various companies, including the Pittsburgh & Butler Railway Co., Butler, Pa., on electrical repair and car equipment work; two engagements with his first employer, the Valley Iron Works, where he was assistant to the superintendent and had charge of electrical equipment and inspection and also compiled and edited a catalog; the Lycoming Foundry & Machine Co., Williamsport, designing special machinery and tools and promoting

their sales; and the Williamsport Gas Engine Works as assistant to the manager. The next four years he was a draftsman and designer with the Ingersoll-Rand Company at Painted Post, N.Y. He returned to his birthplace in 1914 as chief draftsman for the E. Keeler Company. From 1917 to 1919, Mr. Rathmell was chief engineer of the Bryan Harvester Company, Peru, Ind., resigning to become assistant to the manager of the Franklin, Pa., plant of the Chicago Pneumatic Tool Company for one year. This employment was followed by five years with the Dunn Road Machinery Company, Conneaut, Ohio, where, as designing engineer and manager, he had general control of the development and marketing of a concrete road tamper and finishing machine which he invented and on which a patent was granted on February 13, 1923. From 1925 to 1934, Mr. Rathmell was development engineer for the Western Wheeled Scraper Company, Aurora, Ill., and secured patents on several mechanisms pertaining to earth-moving machinery. Illness compelled retirement from active business, but for several months prior to his death, Mr. Rathmell was working on a plan that would enable vehicles to travel on rails or roads without altering the construction of the vehicle. This plan required the fitting of steel flanges on the outside of each wheel immediately above the tire and these flanges would make contact with a metal guide rail that formed part of a concrete or wooden vehicle track which was laid parallel with and closely adjacent to existing railroad tracks.

Mr. Rathmell was elected to associate membership in the A.S.M.E. in 1921. He belonged to the Scottish Rite in the Masonic fraternity and was a Shriner. He was a student of nature and had been a merit badge councilor and a member of the court of honor for the Aurora Area Council of the Boy Scouts of America since 1927.

His widow, Emma (Miller) Rathmell, whom he married in 1910 and a son and namesake, who resides at Williamsport, Pa., survive him.

ROBERT IRWIN REES (1871–1936)

It was the privilege of many people to know General Robert Irwin Rees, for his friends and the things in which he was interested were legion. It is difficult to realize how one man could have exerted such a remarkable influence in so many and such diversified activities, but it is still more remarkable that his colleagues in each of his interests should have had the feeling that he gave his best efforts, his deepest attention, and his extraordinary leadership to the particular project which they shared with him. It is a mark of genius that from his storehouse of experience he could always be depended upon to draw some unique and splendid gift, some faith that sustained others, some insight which gave direction and purpose to the undertaking, some conclusion that was important and which bore the stamp of truth, and in addition to all this, the energy and force to carry the project to its successful conclusion.

Born in Houghton, Mich., on November 9, 1871, he was a member of a leading family in the copper country. His boyhood was normal and relatively uneventful. His early education was acquired in Houghton schools and at the Michigan College of Mining and Technology, where during his undergraduate years he was selected as assistant in the department of physics; at this time, too, he helped the college librarian and thereby developed an early interest in books and reading which he never lost. Upon completing his course in mining engineering, his broadening interests brought him East to study at Harvard, where the lectures of William James undoubtedly did much to influence his philosophy of life and motivate his intellectual interests. Later he enrolled in the New York Law School where his plans were interrupted by the Spanish-American War.

Motivated by the spirit of service and adventure, he enlisted in the army and began twenty-seven years of a notable military career. Within a short time he was commissioned second lieutenant. His first important assignment at the conclusion of the war was in connection with the Philippine Insurrection, where his unexcelled qualities of leadership in establishing harmony contributed much to the rehabilitation of stricken communities. He had by this time determined upon a military career and zealously set about the study of military affairs and his own development. He attended in succession, the Army School of the Line, of which he became a Distinguished Graduate in 1913, and the Army Staff College from which he was graduated in 1914. Detailed a member of the General Staff Corps at the beginning of the World War, he served in the War Plans, Executive, and Operations Division in Washington. When the need for capitalizing the educational facilities of the country became apparent to military leaders, at the suggestion of educators a Committee on Education and Special Training was organized in January, 1918. As chairman of this committee he was then placed in charge of the military and technical training of technicians and mechanics for the Army in educational institutions. This organization later developed into the Students' Army Training Corps, with the wider objective of preparing college students inducted in the service for commissions in the Army.

The termination of the World War in the signing of the Armistice brought the new problem of providing organized activities for the men in France until they could be brought home, and in December, 1918, he was selected for General Pershing's staff at General Headquarters, at Chaumont. With characteristic rapidity and efficiency, he organized the educational facilities of the Allied forces and in a short time was supervising the educational activities of 287,000 men, and providing lecture courses and institutes for 1,600,000 more.

So outstanding had been his contributions that in February, 1919, he was awarded the Distinguished Service Medal "for exceptionally meritorious and conspicuous service" to the Government. In April of the same year he was made an Officer of the French Legion of Honor; he was the only officer of the United States Army to receive this decoration from the Minister of Education. He was further honored for his signal service in maintaining the morale of the men in France by being made a Citizen of Beaune.

On returning to Washington he was reassigned to the War Plans Division as Chief of the Educational and Recreational Branch. In the few remaining years of his military career he became a graduate of the Army War College in 1923, contributed effectively to the organization of the Veterans Bureau and at the time of his resignation in 1924 to become assistant vice-president of the American Telephone & Telegraph Co., he was a member of the War Department General Staff, Operations and Training Division, in charge of the Reserve Officers' Training Corps, War Department, a member of the Federal Council of Citizenship Training, and a member of the National Research Council.

To his new position on the personnel staff of America's largest business organization, he brought a richness of experience and understanding of education and the problems of induction and adjustment of young men to business that quickly crystallized the purpose and method in the recruitment and training of thousands of young graduates who entered the Bell System during the thirteen years of his connection with it. His perception of and keen interest in educational problems, his unfailing ability and willingness to aid his associates in every possible manner, coupled with a sympathetic understanding and a rare quality of leadership, made many look to him with high regard and appreciation for his counsel and assistance. His understanding of their needs caused many outside organizations to turn to him for cooperation in their work. The two societies in which he was most active were the Society for the Promotion of Engineering Education, of which he was the only industrial member ever to be elected to the presidency, and The American Society of Mechanical Engineers. He was also a member of the American Institute of Mining and Metallurgical Engineers and the American Association for the Advancement of Science, and a member of the executive committee of the American Association for Adult Education and of the Adjustment Service of New York, vice-chairman of the Engineers' Council for Professional Development, and chairman of the executive committee of the National Occupational Conference.

General Rees became a member of the A.S.M.E. in 1929, during his presidency of the S.P.E.E., and was greatly interested in bringing about a closer relationship between the S.P.E.E. and the professional groups. The Fiftieth Anniversary of the A.S.M.E. in 1930 provided a notable occasion for an address by him before the Society. Speaking at the welcoming dinner in New York on behalf of the educational institutions of the country, General Rees concluded as follows:

"The American Society of Mechanical Engineers and other national engineering societies, together with the S.P.E.E., can contribute much in a cooperative effort to meet the challenge of engineering education and practice. The American Society of Mechanical Engineers has recognized its responsibility in generously supporting the summer schools for engineering teachers, and through its Committee on Economic Status has outlined a program which will contribute to the solution of some of the problems mentioned. Particularly are the members to be commended for their efforts in encouraging the continuing intellectual development of young engineers, and in raising to further high standards the profession of engineering. Such cooperation fortifies our brother-members in engineering education in the discharge of the sacred trust which is their inheritance—a trust which, in the development of the young manhood of the nation, guarantees leadership which will make sure the foundations of our American civilization."

In December, 1930, General Rees was appointed a member of the A.S.M.E. Committee on Meetings and Program. He was chairman of the committee in 1934–1935, the last year of his term, and then continued as advisory member of the committee the following year. He was named representative of the committee on the Advisory Board on Technology when it was formed in December, 1934. In 1935 he was appointed to the Executive Committee of the Management

Division and he was one of its representatives on the National Management Council, which he served as vice-chairman. In the work of all these committees he served with distinction, showing fine imagination and splendid judgment.

In recognition of his achievements and contributions to education, he was elected an honorary member of the engineering fraternity, Tau Beta Pi, and was awarded the honorary degree of Doctor of Engineering by Worcester Polytechnic Institute in 1930 and by Michigan College of Mining and Technology in 1933. His position in the field of education was unique; he had no affiliation with any college other than his membership on the Corporation of the Polytechnic Institute of Brooklyn, and yet he was regarded as one of the most active and influential educators in America. Likewise, in the profession of engineering, his vision of the necessity for cooperation in activities which affected the life and development of young engineers was one of the greatest factors in the organization of the Engineers' Council for Professional Development, to which he planned to devote the last years of his active career.

As vice-president and chairman of the Committee on Professional Training, of E.C.P.D., he looked forward jubilantly to the time when he could devote his full energy to this project. It was also his long-time ambition to put into writing some of the convictions and conclusions emanating from his own experience, keen observation, and grasp of essentials. The denial of this opportunity is a tragedy to education, although in more than a hundred addresses and papers recorded in the proceedings of many societies and magazines and in the book "Personnel Management," published by the Alexander Hamilton Institute for its course on Modern Business, there are glimpses of the depth and scope of his thinking. Death came to him on November 23, 1936, less than a month after his sixty-fifth birthday, while he was in Detroit to deliver a talk to a group of engineers. He was buried with military honors in Arlington National Cemetery on November 30, the day on which he was to have been retired from the American Telephone & Telegraph Co. In his passing the professions of education and engineering have lost one of their truest friends and advisers, although the influence of his ideals and aspirations will still be felt as the work which he started on such a firm foundation continues through the efforts of his loyal and devoted friends. [Memorial prepared by Ovid W. Eshbach, American Telephone & Telegraph Co., New York, N.Y.]

WILLIAM J. REILLY (1860–1934)

William J. Reilly, consulting mechanical engineer of Denver, Colo., died of pneumonia in Chicago, Ill., on May 1, 1934. He retired as active manager of the Denver office of the Babcock & Wilcox Co. in May, 1926, after having been associated with the company since 1896.

Mr. Reilly was born at Pollack Shaws, Scotland, on May 2, 1860, the son of Peter and Susanna (Fitzsimmons) Reilly. He prepared for college at St. Mungo's Academy in Glasgow, and took an academic course at St. Vincent College, Latrobe, Pa., after coming to the United States. He served an apprenticeship with Wm. Wood & Co., textile manufacturers, Philadelphia, Pa., from 1875 to 1878 and secured his early shop experience with that company. In 1887 he became superintendent of electrical and steam installations for the North American Construction Company. After three years in that position he went to Evanston, Ill., where he spent three years as manager and engineer for the Evanston Electric Illuminating Company. From 1892 to 1895 he was connected with the Westinghouse Electric & Manufacturing Co., the first two years as superintendent of the installation and operation of the outdoor incandescent lighting at the Columbian Exposition in Chicago. In 1895 and 1896 he was in charge of the Electrical Department of the Terminal Railroad Association, St. Louis, Mo. He then became associated with the Babcock & Wilcox Co., New York, with which he remained until 1926, serving first as superintendent in charge of the Erection Department in New York for two years. In 1901, after spending three years in the Sales Department, he was made manager of the Cleveland office. He was transferred to the Denver office ten years later.

Mr. Reilly was active in the early development of steam power and the smokeless burning of Colorado lignite fuels. He became a member of the A.S.M.E. in 1901 and also belonged to the Denver Athletic Club. He was married in 1889 to Mary Gertrude Morgan, who died in August, 1934. Their children, Rosa Adele, Louis G., and A. G. Reilly, of Denver, and Dr. William J. Reilly, of Chicago, survive them.

JOHN SHELTON ROBINSON (1874–1936)

John Shelton Robinson was born at Huntsville, Texas, on January 8, 1874. He was graduated from Alabama Polytechnic Institute with a B.Sc. degree in 1893. For several years he was employed by the Southern Bell Telephone & Telegraph Co., at Atlanta, Ga., on telephone plant construction and testing. Then followed a long period (1897–1925) with the Columbia Water & Light Co., Columbia, Tenn., which advanced him to the position of manager. His duties covered design and construction to meet the growth and development of the properties, as well as operation. In 1925 he was engaged by the Louisiana Research Corporation of Monroe, La., to carry on research work in designing, supervising, and operating equipment employed in chemical research, and continued in this work for about three years. In 1925–1926 he was also connected with the Southern Cities Power Company of Chattanooga, Tenn., as chief engineer, designing electric transmission lines, hydroelectric and steam plants, and supervising the construction of hydroelectric plants. During 1928–1929 he was engaged by Mack & Collins, of Monroe, to make a survey of New Orleans and vicinity and to prepare estimates on gas equipment and consumption of industrial plants in that area. Since 1929 he had been employed by the Memphis Power & Light Co., and at the time of his death on June 26, 1936, was with the Commercial Sales Division of the company.

Mr. Robinson became a member of the A.S.M.E. in 1930 and was active in Society work in Memphis and a member of the Engineers' Club there.

GEORGE OTIS ROCKWOOD (1872–1935)

George Otis Rockwood, president of The Rockwood Manufacturing Company, Indianapolis, Ind., for 26 years, died in that city on July 2, 1935, from embolism following an operation. He was born at Chattanooga, Tenn., on August 7, 1872, son of William and Margaret (Anderson) Rockwood. He prepared for college in the elementary and high schools of Indianapolis, studied mechanical engineering at Purdue University for two years, and supplemented this by special instruction.

In 1894, he became associated with his father in The Rockwood Manufacturing Company, which the latter had formed some years before to manufacture paper pulleys, friction drives, and transmission specialties. He was superintendent, engineer, and general manager of the company and became its president in 1909 when his father died. Mr. Rockwood held this office continuously until his death and was also president of General Fibre Products, Inc., of which the Rockwood Company formed a division. In his work, he placed friction-drive transmission on a practical engineering basis and did considerable special work in connection with belt drives. Mr. Rockwood, probably more than any other individual, was responsible for the development and introduction of paper or fiber pulleys as an improvement over wood and metal pulleys for the transmission of belt power. A number of patents for the manufacture of, and improvements in, paper and fiber pulleys and associated power-transmission equipment were issued to him.

Mr. Rockwood became a member of the A.S.M.E. in 1911. He was a prominent member of the National Metal Trades Association, National Founders Association, and National Industrial Conference Board and was also a civic leader, being active in city, state, and national business affairs.

Surviving Mr. Rockwood are his widow, Marie (Rich) Rockwood, and a daughter, Diane.

JOSEPH SHIRK RUBLE (1877–1935)

Joseph Shirk Ruble, supervisory projects manager for the Federal Emergency Housing Corporation in the Cleveland district, died at Preston Springs Sanitarium, Preston, Ont., on September 4, 1935. His widow, Carolyn (Newbold) Ruble, whom he married in 1913, survives him.

Mr. Ruble was born at Center Hall, Pa., on November 1, 1877, son of James and Linda (Shirk) Ruble. He was graduated from Pennsylvania State College with a B.S. degree in 1901. For a year after graduation, he worked at the Carrie furnace of the Carnegie Steel Company at Rankin, Pa., and was with the C. H. Veeder Mfg. Co., Hartford, Conn., as mechanical draftsman and designer for another year. From 1903 to 1906, Mr. Ruble was employed in the engineering department of the American Steel & Wire Co., Cleveland, Ohio, where he designed three furnaces and later supervised their construction. Upon the completion of these furnaces, he went with Hoover and Mason, Chicago, Ill., and spent the next two years in charge of the erection of an unloader and bridge for the Pennsylvania Railroad at Ashtabula, Ohio. From 1908 to 1913, Mr. Ruble was superintendent of construction for the Tennessee Coal, Iron & Railroad Co., Birmingham, Ala. The next nine years were passed in the employ of the Samuel Austin & Son Co., Cleveland, as superintendent of construction and later as vice-president. In this connection, he contributed largely, through his tireless energy and rare

ingenuity in the handling of the rapid construction of factory buildings, to the munition production program of the World War. In 1922, Mr. Ruble resigned as vice-president of the Austin Company to become executive vice-president and general construction manager of the H. K. Ferguson Company, Cleveland, and remained there until the dissolution of the company ten years later. While connected with the Ferguson organization, he spent considerable time in the Orient and was in Japan during the 1923 earthquake. From 1932 to 1934, Mr. Ruble was engaged in business at Cleveland on his own account as a contractor and engineer and was then appointed to the position of supervisory projects manager for the Federal Emergency Housing Corporation in the Cleveland district, holding it until his death a year later.

His indomitable energy, rare engineering and executive ability, tireless devotion to business, and personal and technical integrity were known throughout the building industry not only in the United States but also in the Orient, where he supervised a number of large construction programs.

Mr. Ruble was elected to membership in the A.S.M.E. in 1913. He also belonged to the Associated General Contractors of America, and to the Shaker Heights Country Club and the Cleveland Athletic Club, was a Knight Templar and a Shriner, and a member of the Phi Kappa Sigma fraternity.

FREDERICK A. SCHEFFLER (1858–1937)

Frederick A. Scheffler, a member of the Society since 1883, died at his home in Glen Ridge, N.J., on February 24, 1937. He had long been prominent in the electric and mechanical engineering fields, being one of the original group of Thomas A. Edison's coworkers. He was a charter member and an ardent worker for the Edison Pioneers. For a time he was its treasurer and later its president, after which he reassumed the treasurership, which he held to the time of his death. He was one of the originators of the movement that finally resulted in the organization of the Thomas Alva Edison Foundation, of which he was a director and vice-president.

Mr. Scheffler was born at Galion, Ohio, on December 20, 1858, a son of Theodore and Mary E. (Lewis) Scheffler. He served a year's apprenticeship at the Grant Locomotive Works, Paterson, N.J., then took a course of mechanical studies at the Paterson Seminary, from which he was graduated in 1875. Following his first business experience with the Long Island Railroad, Long Island City, and with the Rhode Island Locomotive Works, at Providence, for each of which he worked as a draftsman for about two and a half years, he spent a year as draftsman for William A. Harris, builder of the Harris-Corliss engine. He also did some part-time work for the Brown & Sharpe Mfg. Co. while in Providence. He then secured a position with the Edison Company for Isolated Lighting, New York, taking charge of the engineering department under Miller F. Moore, general manager of the company. His first assignment was on plans for Edison's second electric locomotive which was tried out on the experimental tracks at Menlo Park, N.J. The trucks and framework are still on display in front of the Edison laboratory at Orange, N.J.

Later he designed the isolated lighting installation for the W.H. Vanderbilt residence on Fifth Avenue, New York, and that of the elder J. P. Morgan on Madison Avenue. Niblo's Garden, a famous amusement place half a century ago, was also equipped with an isolated plant of Mr. Scheffler's design.

In May, 1884, he resigned to accept the position of superintendent of The Erie City Iron Works, Erie, Pa., and after five years moved on to the Westinghouse Electric & Manufacturing Co. at Pittsburgh as acting general superintendent. Changes followed to include affiliations with the Brush Electric Company, Cleveland, Ohio, as general superintendent; The Stirling Company, New York, as general sales manager; and as general factory manager for the Sprague Electric Company, Watsessing, N.J. He then entered the contracting business with Jas. Beggs & Co., New York, and later was general manager of the Marine Engine & Machine Co., Harrison, N.J.

He entered the employ of the Stirling Boiler Company, later the Stirling Consolidated Boiler Company, in 1905 as a sales engineer. Upon the consolidation of Stirling Consolidated Boiler Company with The Babcock & Wilcox Co. he entered the employ of the latter. He was attached to its New York sales office and served with the company until December 31, 1918, when he resigned to enter the employ of Fuller Lehigh Company, manufactures of pulverized fuel equipment, etc. That company was purchased by The Babcock & Wilcox Co. in the early part of 1926 and in that way Mr. Scheffler returned to The Babcock & Wilcox Co. For a long period of years and until his retirement in 1927 he was an exceptionally successful salesman.

He was an active member of The American Society of Mechanical Engineers, presenting papers and taking part in the discussions. Among the papers which he presented were the following:

A New Method of Constructing Horizontal Tubular Boilers (Trans., 1885)

Tests of a Boiler Using Grates With Small Percentage of Openings (Trans., 1893–1894)

The Potter Mesh Separator and Superheater (Trans., 1902)

Suggestions for Shop Construction (Trans., 1904)

Pulverized Coal for Stationery Boilers (with H. G. Barnhurst) (Trans., 1919)

He was a member of the House Committee of the Society for five years (1913–1917) and its chairman in 1917. He had served on the Dinner Committee for the Annual Meeting continuously since 1925, his being the willingly accepted responsibility of special arrangements for the groups of thirty-five, forty, forty-five, and fifty-year members. It was a particularly happy occasion for him when he was presented with the fifty-year badge of membership at the Annual Dinner in 1933. He was a recognized leader of the old guard that has done so much to add to the prestige of the Society.

He was a Fellow of the American Institute of Electrical Engineers. He served the Institute as a member of the committee on power stations from 1916 to 1919 and again from 1922 to 1935, and as a member of the special committee on the Edison Memorial during 1932 and the Edison Medal Committee, 1913–1918, 1922–1927, and 1930–1933. He was a past-president of the New York Electrical Society, and a member of the Lawyers' and Engineers' clubs of New York. He served several terms as alderman on the Glen Ridge Borough Council and was a member of the Glen Ridge Congregational Church, the Glen Ridge Country Club, and the Senior Golf Association of Essex County. He had been interested in golf for many years and on occasion played as many as thirty-six holes a day. As recently as the summer prior to his death he was sufficiently active to continue to play.

Mr. Scheffler married Miss Linda Rose, of Passaic, N.J., in 1885, and is survived by her and by a daughter, Lucille (Mrs. Robert T. Lozier).—[Memorial prepared by D. S. Jacobus, New York, N.Y. Mem. A.S.M.E.]

JULIUS AUGUST SCHWANTES (1880–1936)

Julius August Schwantes, mechanical engineer in charge of mechanical research for the Colgate-Palmolive-Peet Company, Jersey City, N.J., died on December 27, 1936. He had been with this company, and its predecessor, The Palmolive Company, for twenty years.

Mr. Schwantes was born at Two Rivers, Wis., on February 18, 1880, his parents being William and Emma (Kortens) Schwantes. He was graduated from the Two Rivers High School in 1897 and during the next two years taught school, at the same time taking a course in mechanical engineering from the International Correspondence Schools. He served an apprenticeship with the Hamilton Manufacturing Company at Two Rivers, worked there as shop foreman until 1906, and after six months' experience in the machine shop in the water turbine division of the Allis-Chalmers Manufacturing Company, became assistant works engineer for the Hamilton company. During the three years he held this position he spent some time on design, and developed special woodworking machinery.

His next position was that of works engineer for the Willow Grass Rug Company, at Green Bay, Wis. Here he designed, built, and patented a grass twine machine and installed some forty of them. He also changed the entire plant from direct to alternating current, and supervised the building of additional buildings.

Mr. Schwantes entered the employ of The Palmolive Company in November, 1916, as works engineer. In his application for membership in the A.S.M.E. in 1925, he reviewed his first ten years with the company as follows:

"During my employment here I have designed and patented a number of laborsaving devices and machines, among which is a plodder with an automatic cutting device, an automatic sealing machine, automatic soap-bar machine, new soap-boiling method, shaving-stick holders, and many other devices. I have charge of the mechanical engineering in all our branches. At present I am assisting in laying out a factory in Edgewater, N.J. I have had direct supervision in designing and building additions to our Milwaukee factory, and of the installation of boilers, engines, large tanks and kettles. Some of these tanks and kettles are 20 in. in diameter by 40 in. deep and were installed inside the old building while the upper floors were kept in full operation."

The name of the company was changed following a merger in 1928 and since that time Mr. Schwantes had engaged largely in mechanical research.

Mr. Schwantes had been a member of the National Association of

Stationary Engineers (now the National Association of Power Engineers) and of the Engineers Society of Milwaukee. He is survived by his widow, Laura E. Schwantes, whom he married in 1909, and by their son, J. Warren Schwantes.

GEORGE EDWIN SEABURY (1865–1936)

George Edwin Seabury, since 1917 superintendent of the Station Engineering Department of The Edison Electric Illuminating Company of Boston, died on July 13, 1936, in New Haven, Conn., of cerebral embolism. He is survived by his widow, Emma Augusta (Hodgdon) Seabury, whom he married in 1891; a daughter, Ruth Isabel Seabury, residing with Mrs. Seabury in Jamaica Plain, Mass.; and three sons, Dr. Robert Brewster Seabury, New Haven, Conn., Major Alden Humphrey Seabury, U.S.A., stationed at Monterey, Calif., and Gordon Hodgdon Seabury, who is with the Edison company.

Mr. Seabury was born at Yarmouth, Me., on August 6, 1865, son of Sumner and Sarah Elizabeth (Humphrey) Seabury. He attended the Fort Fairfield (Me.) High School and was graduated from the University of Maine in 1888 with a B.M.E. degree. His experience prior to his connection with the Edison company included two years as draftsman and erector with the Maine Central Railroad at the Waterville shops; six years with the Hinkley & Eglery Iron Co., Bangor, Me., successively as draftsman, foreman of the pattern shop, and assistant superintendent; two and one-half years with the International Paper Company, Livermore Falls, Me., during which time he progressed from the position of draftsman to that of master mechanic; four years with the Edison Portland Cement Company, at Stuartsville, N.J., as general master mechanic of construction; two years with Rathbone Sard & Co., Albany, N.Y., in the position of general factory superintendent, manufacturing stoves and gas ranges; and six years with the General Electric Company at Schenectady, N.Y., part of the time as foreman of steam-turbine construction and later as assistant foreman of assembly and tests of air compressors.

Mr. Seabury entered the employ of The Edison Electric Illuminating Company of Boston in the capacity of erecting engineer of the Station Engineering Department in 1912 and became superintendent in October, 1917.

A member of the A.S.M.E. since 1916, Mr. Seabury served as chairman of the Boston Section in 1926–1927. He was chairman of the Engineering Societies of Boston in 1930–1931, and was a member of the Engineering Societies of New England and of the Engineers' Club, Boston. He was a past-president of the Boston Alumni Association of the University of Maine, a 32d degree Mason and past-master of the Mt. Lebanon Lodge. He belonged to the Eliot Club and Euclid Associates, and was active in church and related work. He was a trustee of the Boylston Congregational Church in Jamaica Plain, director of the City Missionary Society of Boston, a member of the Congregational Club, and a supporter of the work of the Y.M.C.A., Salvation Army, and similar organizations.

GEORGE CARL SHAAD (1878–1936)

George Carl Shaad, highly esteemed dean of the School of Engineering and Architecture of the University of Kansas, came to an untimely death on July 6, 1936, at Pasadena, Calif., as the result of an intestinal infection. Dean Shaad left Lawrence the day following Commencement by automobile accompanied by Mrs. Shaad and their youngest son, to attend the summer convention of the American Institute of Electrical Engineers in Pasadena. With preparations carefully made for a trip which was to be an outing combined with business, he departed with the enthusiasm of a youth, only to contract the fatal affliction along the way, and arrived in Pasadena already very ill.

Dean Shaad was an outstanding educational and professional leader. At the time of his death he had just completed a term as chairman of the Kansas City Section of the A.S.M.E., of which he became a member in 1914, and he was serving as a director of the American Institute of Electrical Engineers, of which he was a Fellow. He was an active worker in the Society for the Promotion of Engineering Education, of which he was vice-president in 1934. He had been active in the Kansas-Nebraska Section of this society since its organization. He was a member of the State Board of Professional Engineers of Kansas, the examining board for licensing engineers in the State of Kansas. He also had been named as a member of the visiting committee of the Engineers' Council for Professional Development for this region. For the University of Kansas he was serving as its representative on the Board of Faculty Representatives of the Big Six Athletic Conference and was chairman of that board. He contributed numerous articles to the technical press, and his work, "The Construction and Operation of Electrical Power Stations," was published as a section of a standard electrical engineering handbook.

Son of George and Christina (Ernst) Shaad, Dean Shaad was born in Stratford, N.Y., on May 5, 1878. His early education was received at Stratford. He went to The Pennsylvania State College, and received a B.S. degree in electrical engineering in 1900. In 1905 he was awarded the E.E. degree. From 1900 to 1902 he was employed in the testing department of the General Electric Company at Schenectady, N.Y. He began his teaching career in 1902 as an instructor in electrical engineering at the University of Wisconsin; he was promoted to an assistant professorship in 1904. In 1906 he became an assistant professor at Massachusetts Institute of Technology, and was promoted to an associate professorship in 1907. He served in this capacity until 1909; then he was called to the University of Kansas as professor of electrical engineering, where he became head of the Electrical Engineering Department. During 1917–1918, owing to the absence of the regular dean in the military service, he served as acting dean of the School of Engineering, and upon the death of Dean Perley F. Walker in 1927 he was appointed dean. He was married to Miss Merthyr Tydvil Evans in 1906, and is survived by Mrs. Shaad, one daughter, and three sons, two of whom are in the employ of the General Electric Company at Schenectady.

Dean Shaad was a member of Tau Beta Pi, Phi Kappa Phi, and Sigma Xi, honorary fraternities, and of Theta Tau, professional fraternity. He was a member of the Kansas Engineering Society, Kiwanis Club, the Chamber of Commerce, and the Lawrence Country Club. On the golf course he excelled in ability as he did in his profession. He played as energetically and enthusiastically as he worked. His summer vacations were spent mostly in the woods and lakes of northern Minnesota.

There is not a member of the staff of the School of Engineering, nor an alumnus or student of the school who knew him that does not feel that he has lost a friend. Here was a man with simple honesty, a natural dignity, a friendly nature, absolutely informal in his relations with others, and one in whom you felt immediately that you could place your trust. There was never a hesitancy on the part of students to approach him when in trouble, or even when delinquent. They could always feel that they would be treated fairly. He was always ready to be of service to others and would enter enthusiastically into any undertaking which promised benefit to the engineering profession, which he cherished. The loss of a man like this can be softened only by the realization that our lives have been made richer spiritually by associating with him and that the ideals of our profession not only have been upheld but have been uplifted by his example.—[Memorial prepared by John A. King, Professor of Mechanical Engineering, University of Kansas. Mem. A.S.M.E.]

ROY ARTHUR SILENT (1890–1936)

Roy Arthur Silent, chief engineer of the Petroleum Securities Company, Los Angeles, Calif., died on November 11, 1936, at the Good Samaritan Hospital in that city, following an operation performed several days before. He was a native of Los Angeles, where he was born on July 1, 1890, the son of Edward D. and Georgia (Dennis) Silent, and grandson of Judge Charles Silent. He entered the University of California from the Los Angeles High School and took four years of a five-year course there, studying mechanical engineering and irrigation.

After leaving college in May, 1913, Mr. Silent was engaged for something more than a year in the supervision of the erection of a five-story office, loft, and theater building in Los Angeles, including the layout and purchase of mechanical equipment for it, and also assisted during this time in the preparation of technical mechanical data for a protracted law case involving the efficiency of well construction and pumping equipment. With this initial experience he entered the office of J. B. Lippincott, consulting engineer, Los Angeles, and in April, 1916, became an associate member of the firm, with which he continued until the United States entered the World War in April, 1917. His duties included the design and installation of several irrigation and pumping works, the design of gates and structures in the San Fernando District of the Los Angeles County flood-control project, work on the valuation of the property of the Tulare County Power Company and Kern River Water Rights, and several other special assignments involving flow of water, pumping equipment, and construction work.

During the War Mr. Silent served as lieutenant, Class 2, in the U. S. Naval Reserve Force. He was assistant engineer officer of the *U.S.S. Frederick* from April, 1917, to June, 1918, and its senior engineer officer from then until he was ordered to inactive duty on January 1, 1919. He also acted as consultant to the engineering department of the Cruiser and Transport Force.

He returned to civil life as superintendent engineer of the Pan American Petroleum & Transport Co., in charge of the company's work at the home port at Tampico, Mexico. He supervised repair

and maintenance work, installed a work-order and cost-accounting system, and prepared periodic reports on the fleet of tankers operating from Tampico. In 1920 he was transferred to the position of terminal superintendent of the Huasteca Petroleum Company, the producing company for the Pan American Petroleum & Transport Co. He had charge of all docking and shipping, the housing and care of employees, operation of a central hotel, provision for water supply and sanitation, and the maintenance of an industrial plant for construction and repairs. In July, 1924, he took over the entire management of the company's affairs in Mexico and in October of the following year he was assigned the title of manager of the Tampico Terminal Division of the Huasteca Petroleum Company, and appointed representative not only of this company but also of the Mexican Petroleum Company of California, Tuxpan Petroleum Company, Tamihua Petroleum Company, and several others. He represented these companies in the Association of Producers of Petroleum in Mexico and served as chairman of that organization for one year. He was a member of the board of directors of the American School Association, Tampico, and president of the Gorgas Hospital Association, the largest American hospital in the district.

Mr. Silent became engineer for the Petroleum Securities Company in July, 1926, and was assigned to the development of the Doheny Stone Drill Company. Under his direction a million-dollar industrial plant was constructed for that company. Subsequently he developed and marketed oil well drilling machinery and specialties and in 1929 designed and constructed a million-barrel reservoir near Bakersfield, Calif. In addition to his other duties he was engaged in general petroleum engineering, drilling operations, pipe line and maintenance work, supervision of water supply, and civil engineering.

Mr. Silent became an associate-member of the A.S.M.E. in 1922 and a member in 1929. He also belonged to the American Petroleum Institute, Psi Upsilon fraternity, and the California Club, Los Angeles. A number of papers on his experience in the oil industry were published in the technical press. He was greatly interested in photography and in drama, on both stage and screen.

Mr. Silent is survived by his widow, Mary Kate (Dunne) Silent, whom he married in 1914, and by two daughters, Patricia Georgina and Catherine Dunne Silent.

GEORGE PERCY SIMPSON (1868–1935)

George Percy Simpson, who was elected a member of the A.S.M.E. in 1917, was born at Toronto, Ont., Can., on February 24, 1868, son of William and Caroline (Mackenzie) Simpson, and died on April 20, 1935, at Montreal, Que. He prepared for college at the MacTavish School, Montreal, and Trinity College School, Port Hope, Ont. He entered the Massachusetts Institute of Technology with the Class of 1889 but was compelled to leave at the end of the freshman year on account of the death of his father, who was manager of the Canadian Bank of Commerce. He became an apprentice in the locomotive shops of the Canadian & Pacific Railway Co. at Montreal, and subsequently was promoted to chief draftsman, and then to assistant mechanical engineer. After about ten years with that company he became general superintendent of the Dominion Cartridge Company, Montreal, and had full charge of its ammunition department. He resigned in 1904 and came to the United States, where he engaged in some minor mechanical work for a short time, then was appointed consulting engineer of the Robin Hood Ammunition Company, Swanton, Vt. In 1913, Mr. Simpson was made a superintendent of the Oven Equipment & Mfg. Co., New Haven, Conn., and later served as its vice-president for a number of years. During the World War, he acted in an advisory capacity as an expert on munitions. From New Haven, Mr. Simpson returned to Montreal and became vice-president and general manager of the General Combustion Company, Ltd., and also was engaged in the practice of consulting engineering when he died.

Mr. Simpson was interested in machine design and had been granted several patents covering various mechanical devices. He was an ardent sportsman and took keen interest in amateur theatricals.

His widow, Lewena (Chapman) Simpson, whom he married in 1904, survives him, together with a brother, Frank M. Simpson, an engineer with the Bell Telephone Company of Canada, located at Montreal.

ROBERT LEWIS SMITH (1887–1935)

Robert Lewis Smith died at his home in Winchendon, Mass., on August 19, 1935. He joined the A.S.M.E. as an associate member in 1919 and was transferred to the grade of member in 1922.

Mr. Smith was born on September 18, 1887, at Boston, Mass., and was the son of Lewis and Kate (Paul) Smith. He prepared for college at the Mechanic Arts High School of Boston and was graduated in 1909 from the Massachusetts Institute of Technology with the degree of S.B. For the year following his graduation, he served as an assistant instructor of mechanical engineering in its steam and hydraulic laboratory.

All of Mr. Smith's business career was spent in the employ of Baxter D. Whitney & Son., Inc., Winchendon, Mass. In 1910, he accepted the position of mechanical engineer and was in charge of the production of woodworking machinery and tools in its machine shop and foundry for eight years.. He was promoted to factory manager in 1918 and supervised all manufacturing departments until 1923, when he was made general manager. Mr. Smith held this position and also was a member of the board of directors at the time he died. In addition to his duties as factory manager at the Whitney plant, he was engineer for the Winchendon Electric Light & Power Co. from 1911 to 1916 and had charge of the design and construction of its high-tension power station.

Many of the electrical and mechanical devices used on the Whitney line of woodworking machinery were developed by Mr. Smith, probably the most important of these being the electrical brake mechanism for stopping rotary shaper spindles, patented in 1930.

In addition to his A.S.M.E. membership, Mr. Smith was a member of the American Institute of Electrical Engineers from 1919 to 1927. He was prominent in the Masonic fraternity. At the time of his death, Mr. Smith was a selectman of the Town of Winchendon, and he also had served on various committees that were interested in the well-being of the town. He was particularly fond of hunting and golf.

His widow, Madeline (Davis) Smith, whom he married in 1911, and three daughters, survive him.

CLARENCE WILBUR SPROULL (1884–1935)

Clarence Wilbur Sproull, who died at Newport, R.I., on May 31, 1935, was born on August 11, 1884, at Dawn, Ohio, son of Jacob Treber and Sarah Emma (Coppers) Sproull. He spent about a year at the U.S. Naval Academy at Annapolis and was graduated from Rose Polytechnic Institute with a B.S. degree in 1909; ten years later he received an M.E. degree from the Institute.

Following his graduation he worked for a year as erecting engineer for the Connersville (Ind.) Blower Company. He then entered the teaching profession as instructor at Carnegie Institute of Technology, Pittsburgh, Pa., where he remained until 1924. He advanced through the instructional grades of assistant professor and associate professor, and from 1918 until he left Pittsburgh he was professor in charge of the department of drawing and machine design.

From 1924 to 1929 Mr. Sproull served as train control engineer for the Union Switch & Signal Co., at Swissvale, Pa., and during the next few years was chief of design, research department, Champion Coated Paper Company, Hamilton, Ohio. He had been working at the Naval Torpedo Station at Newport since June 1, 1934.

Three patents had been granted to Mr. Sproull for fluid-pressure braking apparatus for controlling the automatic application of brakes on a railway train. He contributed an article on the "Essentials of Cam Design" to the *American Machinist* in 1913.

Mr. Sproull became a member of the A.S.M.E. in 1924, and had been a member of the Engineers' Society of Western Pennsylvania for many years. He was a Mason and his interests included woodwork, music, and firearms. He had made a number of articles from wood, played the violin, and was an excellent marksman. He is survived by his widow, Amy (Leonard) Sproull, whom he married in 1921, and by two children, Natalie Elizabeth Sproull and Lois Marie Sproull, the latter a daughter by his first marriage, in 1909, to Marie Williams, who died in 1918.

In April, 1918, Mr. Sproull applied for a commission as captain of ordnance, in the Officers' Reserve Corps, U.S.A., but the Armistice was signed before any action had been taken. In this connection, letters from those familiar with his work at Rose Polytechnic Institute and Carnegie Institute of Technology paid unusual tribute to his scholastic standing, ability as a teacher and engineer, and personal qualities.

FRANK F. STORMS (1864–1935)

Frank F. Storms, who had been associated with the drop-forging industry for nearly half a century and organized two companies for their manufacture, died on June 2, 1935. He was born at Slatersville, R.I., on January 16, 1864. His education was acquired in the schools of West Boylston, Mass.

From 1884 to 1886, Mr. Storms was an apprentice in the shops of George L. Brown & Co., Worcester, Mass., where he learned to operate planers, gear-cutting lathes, and milling machines and gained some experience in toolmaking. He worked as a machinist and toolmaker with the Iver Johnson Company, Fitchburg, Mass., and, in the six

years that he was employed there, learned die sinking and drop forging, the latter of which became his lifework. From 1892 to 1900, Mr. Storms was superintendent and designer of dies for the Springfield Drop Forging Company, Springfield, Mass. He next organized the Page-Storms Drop Forge Company, Chicopee, Mass., and, in 1907, supervised the planning, construction, and placing in operation of its new plant. Mr. Storms was president of the company from its formation until the plant was sold to the Westinghouse Electric & Manufacturing Co. at the time of the World War and continued as president and general manager of the plant for the Westinghouse interests. In 1920, he organized the Storms Drop Forging Company, Springfield, Mass., and he served as its president and general manager until his death.

Mr. Storms had been a member of the A.S.M.E. since 1915.

GEORGE AUGUSTUS SUTER (1862-1935)

George Augustus Suter, for many years a recognized leader in heating and ventilation engineering in this country, died at his home in New Canaan, Conn., on May 30, 1935, after an illness of about six weeks.

Mr. Suter was born in New York, N.Y., on June 17, 1862. He attended public school, and was a student in the College of the City of New York and graduated from the Columbia School of Mines in 1883 with the degree of Mechanical Engineer.

Following his graduation he was employed for a time by Charles T. Porter in his test of the Saratoga pumping machinery and at his office in New York. In 1884 he took a position with the Exhaust Ventilator Company of Chicago, as engineer in charge of designing and installing ventilating equipment. This work determined the occupation he was to follow for the remainder of his career. Two years after his graduation he became manager of the New York Exhaust Ventilator Company and later was employed by Baker, Smith & Co., as an expert in the ventilation of buildings.

In 1892 he established the firm of G. A. Suter & Co., New York, engineers and contractors for steam power, steam and hot water heating and ventilating apparatus. The firm's business was incorporated in 1905 with Mr. Suter as president of the company. He continued in that office until his retirement from active work in 1912.

During this period he was called upon to install heating and ventilating plants in many large buildings in New York. Among them were nine at Columbus University, including Teachers College; the Horace Mann School, the University Club, Cornell Medical College, Tiffany Building, Singer Tower, six Y.M.C.A. buildings, American Surety Company, Chamber of Commerce, Hanover National Bank, Knickerbocker Hotel, and the Hall of Records.

Mr. Suter was frequently called in as a consultant on large building operations, of which the Grand Central Terminal was one.

In 1917, when the United States declared war, he became consulting engineer to the Housing Corporation at Washington, which was engaged in erecting many buildings made necessary by the War.

With the coming of peace, Mr. Suter retired to his home in New Canaan. He nevertheless retained an active control over his many and varied business interests.

In 1886, he married Miss Sara Ballard Hederick of Brooklyn, N.Y. Mr. and Mrs. Suter had six children, of whom four are living—Mrs. Henry A. Miles, of New Rochelle, N.Y., Mrs. Charles Staiger, of Scarsdale, N.Y., and Marguerite and George Suter of New Canaan. His widow died on April 15, 1937.

Throughout his life he devoted much time and energy to philanthropic and civic affairs. He was of a retiring disposition and much of his most useful work in these fields was done without any publicity. He was president of the New Rochelle Hospital Association from 1899 until 1911 and was active in promoting inland mosquito control work as a measure for stamping out malaria, serving on a local committee which operated through private subscriptions and whose work was later carried on by the town with the cooperation of the State Experimental Station.

He was a member of a special committee appointed to study water supply and sewage disposal for the town of New Canaan, and a new plant operated by the Borough was largely inspired by the report of the committee.

Mr. Suter was much interested in the relief problem and prepared food budgets which were the basis of the work of the first Welfare Committee of New Canaan.

He became a junior member of the A.S.M.E. in 1883 and a member ten years later. He had been a member of the Engineers' Club and Columbia University Club, New York, and of the Wykagyl Club, New Rochelle. He was an ardent yachtsman and lover of outdoor sports.—[Memorial adapted from obituary published in *New Canaan* (Conn.) *Advertiser*.]

FREDERICK SAMUEL THOMPSON (1874-1936)

Frederick Samuel Thompson, special agent for the Travelers Insurance Company, Hartford, Conn., died at his home in that city on September 12, 1936, of thrombosis of the heart. Son of Stephen and Emily (Vilas) Thompson, he was born at Exeter, Kan., on October 4, 1874. His mother had been a teacher prior to her marriage and, after he was graduated from elementary school, assisted in preparing him for high school. In addition to this, Mr. Thompson had private instruction and took some International Correspondence Schools courses.

At the age of 17, he became an apprentice at the Vermont Spool & Bobbin Co., Burlington, Vt., and, upon the completion of this apprenticeship in 1895, was a workman, foreman, inspector, and assistant superintendent in various factories for the next five years. From 1900 to 1905, Mr. Thompson designed wood-cutting tools and designed and built turret lathes and tools. He then accepted a position as draftsman in the Pond Works of the Niles-Bement-Pond Company at Plainfield, N.J., and had risen to assistant chief engineer by 1912, when he became chief engineer of the Hartford Machine Screw Company, Hartford, Conn. Mr. Thompson was subsequently elected vice-president of the company and remained there for several years. During the World War, he was fuel conservator and performed other service. In 1929, Mr. Thompson was elected president of the Ridgefield Company and served until 1934, when the company was dissolved.

Mr. Thompson was elected a member of the A.S.M.E. in 1913. He was interested in the civic and religious activities of Hartford, served as chairman of the committee responsible for the construction of the Thomas Snell Weaver High School and the Central Baptist Church, and was a member of the board of trustees of the latter at the time of his death. Mr. Thompson belonged to the Knights of Pythias, the Masonic fraternity, and the Y.M.C.A.

On October 1, 1902, he married Miss Emma Gilbert, who survives him.

HERMAN ALEXANDER TONNE (1881-1937)

Herman Alexander Tonne was born in Buckow, Saxony, Germany, on November 14, 1881, son of Hermann and Minna Luise (Alex) Tonne. Following the completion of his high-school education at Magdeburg he served a year's apprenticeship as patternmaker and machinist with the Edward Bendel Machine Tool Company there, at the same time attending a technical evening school. He then entered the engineering college at Zwickau, Germany, from which he received a degree in electrical engineering in 1902. He continued his studies at the polytechnic institute in Zürich, Switzerland, securing a degree in mechanical engineering there in 1903.

Mr. Tonne came to the United States in the summer of 1903 and obtained work as a machinist for the Westinghouse Electric & Manufacturing Co., in Newark, N.J. He was employed there for a little over a year, then worked for a few months for the Gould & Eberhardt Co., Newark, as mechanical draftsman. In January, 1905, he began a period of ten years with the Stewart Hartshorn Company, roller curtain manufacturers, East Newark, serving successively as draftsman of tools and furnaces, designer of special machinery, and assistant to the superintendent. He worked on metal stampings, machine and tool design, power presses with micro feeds, slitting and forming multiple mills, automatic spring-winding machinery, special assembly equipment, and the rearrangement of operations for low-cost production. During his early years in Newark he took a technical school course, which he completed in May, 1905.

After leaving the Stewart Hartshorn Company in February, 1915, Mr. Tonne was employed successively by the Ball Bearing Shade Roller Company, Naugatuck, Conn., the Bristol Company and Ingersoll-New England Watch Company, both of Waterbury, Conn., and the Standard Shade Roller Company, at Ogdensburg, N.Y., in each case as special designer. In 1918-1919 he was tool designer for the Brown-Lipe-Chapin Company, Syracuse, N.Y., developing tools, gears, and gages for automobile differentials, wartime aircraft parts, and methods for increased production and experimental research. The next two years were spent in New York, N.Y., with A. D. Smith & Co., designing tools and equipment for a new manufacturing plant, and supervising operation. In 1922-1923 he was tool designer for the Intertype Corporation, Brooklyn, and from then until 1926 design engineer for the Linde Air Products Company, New York, in charge of the design of automatic welding machinery, compressors, and acetylene generators. He also planned piping layouts for liquid air and chemical plants.

From 1926 to 1932 Mr. Tonne was designer and checker for the Brooklyn Edison Company, at the Hudson Avenue Power Station. He made equipment and piping layouts, designed structural steel

work, platforms, ducts, fans, pumps, and conduits, and supervised computations and experimental research.

During the last few years of his life Mr. Tonne was connected with the Department of Parks, Brooklyn, serving as assistant field payroll supervisor from December, 1933, to November, 1934, and from then until his death as assistant supervising draftsman (squad chief), in charge of alterations in field design and progress records of all park construction in the Borough of Brooklyn, including park buildings, playgrounds, bathhouses, swimming pools, etc. He was stationed at the Prospect Park Parade Grounds. Mr. Tonne died of heart failure while on his way to work on January 6, 1937.

An associate-member of the A.S.M.E. since 1917, Mr. Tonne was automatically transferred to full membership in 1935. He belonged to the Kings County chapter of New York State Society of Professional Engineers and served on a special committee with representatives of other boroughs regarding the standing and salaries of licensed engineers under the Public Works Administration and similar agencies. He had been a citizen of the United States since 1910. He contributed several articles to the technical press.

Mr. Tonne married Ilona M. Royko in 1907, and is survived by her.

FREDERICK TRANSOM (1879–1934)

Frederick Transom, patent lawyer and engineer, of Washington, D.C., who died in the fall of 1934, was born in Brooklyn, N.Y., on May 18, 1879. From his application for membership in the A.S.M.E., made out in 1921, is taken the following record of his early experience.

He served a four-year apprenticeship with Bement, Miles & Co., of Philadelphia, Pa., at the same time taking evening courses in mathematics and drawing. Shortly after the completion of his apprenticeship he entered the University of Pennsylvania, from which he was graduated with a B.S. degree in mechanical engineering in 1895. Following his graduation he worked for the Standard Oil Company, first as draftsman and later as an operating engineer in charge of a steam power and electric lighting station. He left the company to enter the Patent Office after passing a civil examination. Subsequently he became first assistant examiner in the Patent Office.

The University of Pennsylvania bestowed the degree of Mechanical Engineer upon him in 1906 and in that year he also received the degree of Bachelor of Laws and Master of Patent Law from George Washington University. His work at the Patent Office dealt particularly with electrical cases.

He is survived by a sister, Frances (Mrs. John) Hamman, of Houston, Tex.

WILLIAM STANTON TWINING (1865–1937)

William Stanton Twining was born near Titusville, Pa., on February 20, 1865, the son of Charles and Mary (Stanton) Twining. His parents removed to Union City, Pa., in 1870, where he entered the public schools, from which he was graduated in 1881. He then entered the employ of the Industrial Iron Works, at Union City, as a machinist's apprentice and remained there three years. In September, 1884, he entered Cornell University, at Ithaca, N.Y. A year later, he entered Allegheny College, at Meadville, Pa., from which he was graduated as a civil engineer with the degree of Bachelor of Science in 1887. He remained at Allegheny College as an instructor in civil engineering and chemistry until 1890. He was granted the degree of Bachelor of Arts in 1889, and served as a trustee of the college from 1923 to 1935.

Mr. Twining's long connection with the development of the electric street-railway industry began when he entered the Railway Engineering Department of the Thomson-Houston Electric Company, in Boston, Mass., as assistant engineer, and was associated with the design and erection of steam-driven electric power plants at Indianapolis, Ind., Allentown, Pa., and Toledo, Ohio. In 1891, he entered the employ of the Harlem Bridge, Fordham & Morrisania Railway Co. (later, a part of the Union Railway Company of New York) as principal assistant engineer in charge of the electrification of the street-railway system, and, in the following year, he was engaged on similar work with the Atlantic Avenue Railway Company (later, a part of the Brooklyn-Manhattan Rapid Transit System).

In 1893, Mr. Twining entered the employ of the Peoples Traction Company, of Philadelphia, Pa., as assistant to the chief engineer, and was made chief engineer in 1895. In that year, the principal street-railway lines of Philadelphia were consolidated as the Union Traction Company and Mr. Twining was made chief engineer of the consolidated system.

During this period, the streetcar lines passed from the horse-drawn type to electric operation, and the successful reconstruction of the roadbed, with its overhead trolley system, new cars, large generating plants, with their far-flung distribution system, is an enduring monument to the genius, resourcefulness, and skill of Mr. Twining as an engineer.

In 1902, the Philadelphia Rapid Transit Company was formed, and this company leased the electrified street-railway system, controlled and operated by the Union Traction Company. This company also secured public franchises authorizing it to construct and operate a number of high-speed subway and elevated passenger-railway lines.

With the transfer of the Union Traction System to the Philadelphia Rapid Transit Company, Mr. Twining continued as chief engineer, in which position he directed and supervised the construction and equipment of the Market Street Subway and Elevated Passenger Railway; and a two-track subway in Market Street, extending from the Delaware River to the Schuylkill River, at which point it became a two-track elevated-railway structure of steel and concrete, to the western boundary of the city near Sixty-Ninth Street. This system is six miles in length, and its construction embraced some outstanding improvements over the existing structures of this period.

In 1910, he became associated with the firm of Ford, Bacon & Davis, Inc., of New York, N.Y., as an engineering executive and, in this capacity, on behalf of the firm, studied the operating structure of many of the prominent public-utility systems of the United States, submitting reports and recommendations covering operation, valuation, and improvement of the properties.

In 1912, Mr. Twining returned to Philadelphia as a representative of Ford, Bacon, & Davis, Inc., who had been retained to advise the City of Philadelphia in the formulation of a comprehensive plan for the improvement of the passenger-transportation system and to develop and recommend a program for the construction of a system of high-speed transportation lines by the City of Philadelphia. These studies continued until 1916, at which time Mr. Twining severed his connections with the firm of Ford, Bacon, & Davis, Inc., to become the director of the Department of City Transit of the City of Philadelphia, a position that he filled for a period of eight years.

During this time the Frankford Elevated Railway, a two-track elevated structure of steel and concrete, six miles in length, was constructed and equipped for operation. This line connected with the eastern terminus of the Market Street Subway, and extended to the northeast section of the city, known as Frankford. Also, a beginning was made on the construction of the Broad Street Subway on that part of the subway under City Hall.

In 1924, Mr. Twining retired from the public service and devoted his time to consulting practice, and to the study and development of matters of a lifetime interest. He died on February 8, 1937, at his winter home in Fort Myers, Fla.

He possessed, to a rare degree, the ability to conceive, envision, and plan comprehensive engineering systems and structures; he was a master worker in his chosen field of endeavor; his warm personal charm drew his associates into a close relationship and encouraged them to give to their work the best that was in them.

Throughout his professional career, Mr. Twining was steadfast in his convictions and adhered to his conceptions as to what constituted the right and proper course of procedure. He was loved and esteemed by those who were privileged to be associated with him as coworkers and also by those who were admitted to the closer and more intimate tie of friendship.

Mr. Twining was a member of the Union League, The Franklin Institute, and the Engineers Club of Philadelphia, and the Phi Kappa Psi fraternity. His membership in technical societies included The American Society of Mechanical Engineers, which he joined in 1897, the American Society of Civil Engineers, the American Institute of Electrical Engineers, and the American Academy of Political and Social Science.

Mr. Twining was married in 1893 to Mary H. Wylie, in 1900 to Isabella A. Wylie, and in 1916 to Mrs. Harriet P. Rundell, of Toledo, Ohio, who survives him.—[Adapted from memoir prepared by Charles H. Stevens and Joseph W. Silliman, Philadelphia, Pa. for the Transactions of the American Society of Civil Engineers.]

THOMAS ANTHONY VANDER WILLIGEN (1866–1936)

Thomas Anthony Vander Willigen, director of Humphreys & Glasgow Ltd., of London, and its representative in Belgium, Holland, Switzerland, and France, died in Antwerp, Belgium, on March 27, 1936. He was also a director of the Société de Construction d'Appareils pour Gas a l'Eau et Gas Industriels, Montrouge (Seine), France, and active in both firms at the time of his death.

Mr. Vander Willigen was born on August 5, 1866, at Twello, Holland, son of Volkert Simon Maarten and Susanna Antoinetta (Vander Hell) Vander Willigen.

He secured his early education in Haarlem, Holland, and after coming to the United States attended the Stevens Institute of Tech-

nology, from which he was graduated with a degree in mechanical engineering in 1888. Following his graduation he was employed in the drafting rooms of the United Gas Improvement Company, Philadelphia (for which he also worked on the erection of carbureted water gas plants for a time), the & Calumet Hecla Mining Co., Calumet, Mich, and the Winslow Brothers Elevator Company, Chicago, until 1895. He then became chief draftsman for the Buffalo Engineering Company, where he worked until becoming connected with Humphreys & Glasgow in 1896. He worked on the design and construction of carbureted water gas plants for this company at first, later introducing the manufacture of carbureted water gas in Holland and Belgium.

Mr. Vander Willigen became a junior member of the A.S.M.E. in 1896 and a member in 1904. He was a delegate for the United States to the International Congress of General Mechanics, held at Liége in 1930. He was a Mason, for many years a member of the Brussels branch of "Les Vrais Amis de l'Union," and a member of the reception committee of the Rotary Club, Brussels.

Surviving Mr. Vander Willigen are his widow, Helena (Brown) Vander Willigen, whom he married in 1898, a son, Thomas Anton Vander Willigen, of Brussels, and two married daughters, Vera Helena Haigh, Brussels, and Dorothy Violette Kohler, Antwerp.

WILHELM von OSWALD (1859–1936)

Wilhelm von Oswald, privy commercial counsellor and retired assessor of mines, died on March 22, 1936, at Gross-Burgwedel, near Hanover, in Germany. His name recalls the mighty development of the Lorrainese iron industry during the prewar period.

The first generation of the great developers of German industry devoted their labor chiefly to the Rhenisch-Westphalian district. It fell then to the lot of the second generation to create the iron industry in the Lotharingian district. Von Oswald was one of the most outstanding men in that second generation; we may consider him the actual originator of the metallurgical works of Rombach. Originally planned and begun as a blast-furnace plant, Rombach was changed by him into one of the largest and most important metallurgical works.

Wilhelm von Oswald was descended from an old family of jurists. He was born at Strasburg (Western Prussia) on December 18, 1859, and during youth shared the roving life of his father, who was transferred from Strasburg to Brieg and again from there to Danzig. Permanent domicile was only attained by his father upon becoming privy counsellor of the superior court and president of the district court in Arnsberg. There the classical education of Wilhelm von Oswald was completed. At that time ore mining was still in a thriving condition in that region, which circumstance motivated the young man to choose mining for his career. He served an apprenticeship in the government coal mines at Ramsbeck and later took scientific courses at the universities of Leipzig, Berlin, and Bonn. He passed his examination as mine "Referendar" on July 31, 1885, and that of mine "Assessor" on March 15, 1891. He then worked until 1895 as chief of the mining department at the head office of mining authorities at Halle.

In the meantime he married Emma Spaeter, the daughter of Commercial Privy Counsellor Carl Spaeter, who owned the firm of Carl Spaeter at Coblenz, which was at that time the leader in the ore and iron market of Germany. On January 1, 1895, von Oswald became a partner in that firm. Thereby he came in contact with the Rombach Metallurgical Company, the shares of which were, to a considerable extent, held by Carl Spaeter. From 1896 until his death, von Oswald was one of the directors of the Rombach concern, holding office as chairman of its board from 1902 until 1927. He had charge of the plant and of its extension and, in addition to knowledge as a technician, entered that position with broad understanding of economic necessities obtained during his activity in the merchant firm of Carl Spaeter. His vision became enlarged by journeys made in Germany and abroad. The developments which took place at Rombach were frequently new and novel at that time and were often received with antagonism—yet they proved themselves in splendid manner.

By merging the Mosel works and by purchase of large mining fields he increased the company's ore resources. The construction of a coke-oven plant at Zeebrügge gave the works some independence in its supply of coke. Flourishing settlements came into existence all around the works, such as Stahlheim, Rombach, and others. Blessings without end seemed to radiate from these works within the entire district of the Lorrainese. Following protracted negotiations, community of interest was brought about at the beginning of 1914 between the Rombach Metallurgical Works and Concordia Mining Company Limited, with a view to a complete merger of these two enterprises. By that means the coke supply of the Rombach Metallurgical Works was assured, giving them a foundation which seemed sufficiently strong to weather all storms.

The unfortunate ending of the World War dashed Wilhelm von Oswald's lifework to pieces. The beautiful Lotharingian works were soon placed under an administrator and sold to a French group. Fruitless battling of many years to secure an adequate compensation, together with the disastrous conditions of the period of inflation, filled the eventide of this meritorious man's life with great bitterness.

Yet his courage never waned. He knew how to revive former connections of his mercantile firm in foreign parts, which had been torn asunder by the World War. Up to his last days he gave great and efficient care to the affairs of the firm of Carl Spaeter and to those firms of iron merchants which were closely associated with it. Together with the Concordia Company he became active in the field of trading in fertilizers and chemical products and also in the manufacture of sulphuric acid and superphosphate, both in Germany and outside of it.

The German Iron and Steel Institute lost in Wilhelm von Oswald one of its oldest and most noted members. It could always count on his support and assistance, especially as a member of its board of directors, to which he belonged from 1904 until his death.

A rich life came to its close when he died. The name of Wilhelm von Oswald will always live in the history of Germany industry and the German iron trade.—[Translated by Arthur J. Herschmann, New York, N.Y., Mem. A.S.M.E., from tribute published in *Stahl und Eisen* (Zeitschrift für das Deutsche Eisenhüttenwesen), April 2, 1936, p. 432. Wilhelm von Oswald had been a member of the A.S.M.E. since 1899. Surviving him are his widow, two sons, Willi and Kurt, and a married daughter, Eleanor.]

GARDNER TUFTS VOORHEES (1869–1937)

Gardner Tufts Voorhees, widely known as a refrigerating engineer, died on March 18, 1937, after a short illness from typhoid fever, which he contracted while at his winter home in Palm Beach, Fla. With him in Palm Beach were his wife, Ninette (Chretien) Voorhees, whom he married in 1910, and his sister, Mrs. Sydney Montgomery Arnold, and Captain Arnold.

Mr. Voorhees was born on July 6, 1869, in Stamford, Conn., his parents being Abraham and Caroline C. (Tufts) Voorhees. He secured his early education in schools in Somerville and Cambridge, Mass., then attended Allen's English and Classical Preparatory School, in West Newton. He was graduated from Massachusetts Institute of Technology in 1890 with an S.B. degree. During his early years in engineering work Mr. Voorhees was employed by D. H. & A. B. Tower, hydraulic engineers, Holyoke, Mass.; Ludlow (Mass.) Manufacturing Company; and the West End Street Railroad Company, Peabody & Sterns, Steel Cable Engineering Company, and Westinghouse, Church, Kerr & Co., all in Boston. For some years he was connected with the Quincy Market Cold Storage Company, Boston, for which he installed the major part of the brine pipe line that now cools the market district.

Mr. Voorhees had been in business for himself since about 1900. The cooling system installed by him at the Lowney chocolate factory was one of the first air-circulating systems. He installed refrigerating and ice plants for the Boston Athletic Association, the Massachusetts General Hospital, the U.S. Hospital Ship, *Bay State*, the Cincinnati Ice Manufacturing & Cold Storage Co., the Baltimore Cold Storage Company, and many others. Among those whom he served as consulting engineer were the Henry Vogt Machine Company, Louisville, Ky., Ice & Cold Machine Co., St. Louis, Mo., York Manufacturing Company, York, Pa., and Halstead & Co., New York.

He was chairman of refrigeration for the World's Fair at St. Louis in 1904, and was the delegate of the United States to the first International Conference on Refrigeration in Paris, and the second in Vienna.

Mr. Voorhees was the author of a number of books on refrigerating engineering, including two on the absorption refrigerating machine (one on elementary and the other on advanced theory and practice), "The Compression Refrigeration Machine," "Indicating the Refrigerating Machine," and "Refrigerating Machines–Compression, Absorption." His patents, which were numerous, covered basic improvements in the processes and apparatus involved in the production and use of mechanical refrigeration, as well as new processes and equipment for the use of electricity, and developments in thermodynamics and for aircraft. His multiple effect compressor, for increasing the capacity and economy of refrigerating systems, was of particular importance.

Mr. Voorhees had been a member of the A.S.M.E. since 1900. He became a member of the Special Committee on a Standard Tonnage Basis for Refrigeration when it was formed in 1903 and served continuously on it and its successor, the Special Committee on Refrigera-

tion, until 1918. In 1920, shortly after the formation of the Power Test Codes Individual Committee on Refrigerating Machines and Plants, he was appointed a member of it and he served continuously on this committee (now known as the Individual Committee on Refrigerating Systems), until his death.

He was a charter member of the American Society of Refrigerating Engineers, an honorary member of the Southwestern Ice Manufacturers Association and the Western Ice Manufacturers Association, and a member of the Southern Ice Exchange. He belonged to the Salmagundi and Engineers' clubs, of New York, and was a 32d degree Mason.

EDWARD LATHROP VOY (1873–1937)

Edward Lathrop Voy was born on November 22, 1873, in New York, N.Y., son of William H. and Amelia (De Bow) Voy. His schooling was completed in San Francisco, Calif., where he attended high school and studied mathematics and drawing in evening classes, and for about eight years, beginning in 1898, he was connected with the Risdon Iron Works, in San Francisco, doing mechanical drafting for gold dredging and mining machinery.

In the years following the termination of his work for this company, up to the World War, Mr. Voy was engaged in the design, erection, and repair of dredging and excavating machinery for companies in the Middle West and East. These were, in succession, the Allis-Chalmers Company, Milwaukee, and Bucyrus Company, South Milwaukee, Wis., 1905–1908; Stewart-Kerbaugh-Shanley Company, Brewerton, N.Y., on a barge canal contract, 1908–1909; American Locomotive Company, Paterson, N.J., 1909–1911; Furst Clark Construction Company, of Baltimore, Md., on their Cape Cod Canal contract at Sandwich, Mass., 1911–1914; Aluminum Company of America, Massena, N.Y., 1914; Davison Chemical Company, Baltimore, Md., 1915–1916.

Mr. Voy enlisted in Company A, Depot Battalion, 7th Infantry, National Guard of New York. He was transferred to Company E, 7th Infantry, in October, 1917, and honorably discharged in the grade of private in November of that year. Subsequently he served in the U.S. Railroad Administration as engineer in connection with the Cape Cod and New York Canal. He was inspector for the Winnisimmet Shipyard, Inc., East Boston, in 1920, and later engaged in dredging work in Florida.

In 1924 Mr. Voy worked for a time as mechanical engineer for the Burnham Safety Razor Company, New York, and from September, 1924, to January, 1926, he was layout man in the Mechanical Department of Thomas E. Murray, Inc., New York. This was his last active participation in engineering work in the East. He returned to San Francisco in August, 1928, and resided there until his death on February 26, 1937. During these years he did some drafting work for C. C. Moore & Co. and Peak & Co.

Mr. Voy became an associate-member of the A.S.M.E. in 1916 and was automatically transferred to full membership in 1935. He was a Mason and an Odd Fellow and active in fraternal and engineering society work in San Francisco. He is survived by his twin brother, D. C. Voy, of San Francisco, and also by another brother, Charles, and a sister, Florence.

JOSEPH HARRISON WALLACE (1869–1936)

Joseph Harrison Wallace, practically all of whose life was devoted to the pulp and paper industry, died suddenly on July 7, 1936, at Hamilton, Ohio, where he had served since 1931 as consulting engineer for The Black-Clawson Company and the affiliated Shartle Brothers Machine Company, of Middleton, Ohio.

He was the son of Osborne H. and Martha L. (Robbins) Wallace, of Worcester, Mass., where he was born on November 10, 1869, and where he was educated. He received a B.S. degree from Worcester Polytechnic Institute in 1892 and a C.E. degree seven years later. Following his graduation he spent two years with the U.S. Lighthouse Establishment. Working under Major William R. Livermore, U.S.A., lighthouse engineer for the First and Second Lighthouse Districts, in New England, he was in charge of surveys of Government property and of experiments to improve the fog signal service, and prepared a report for the Lighthouse Board on various types of signals. He began as second assistant surveyor but was soon promoted to first assistant surveyor. It may have been this early experience which made him an enthusiastic sailor and yachtsman. He studied naval architecture as a pastime, took great interest in cruiser races, and won for himself some forty cups in three years with a boat which he remodeled. He was commodore of the New York Athletic Club.

In 1894 Mr. Wallace went to Springfield, Mass., to take patent examinations. He stayed with friends in Holyoke, saw the paper mills there, and decided that he wished to work in that industry. He succeeded in securing a job with A. B. Tower & Co., his first assignment being the design of an acid system for a 50-ton sulphite mill. Having no knowledge of the requirements, he visited the Burgess Sulphite Mills at Berlin, N.H., obtained the necessary information, and was able to complete the assignment to the satisfaction of Mr. Tower. He worked as draftsman and afterwards as resident engineer on the construction of paper and pulp mills until the fall of 1897, when he and Mr. Tower formed the partnership of Tower and Wallace, mill and hydraulic engineers, New York. This association continued until the first of February, 1901, during which time the firm designed and constructed many mills in New England, New York, the Middle West, and Canada. The office and drafting room were under the supervision of Mr. Wallace, who also took part in some of the outside work. The driving of a paper machine with a variable-speed twin engine with cranks quartering was applied by him in 1898 to a mill for the Peninsula Paper Company at Ypsilanti, Mich., and the idea was patented by the company, although Mr. Wallace later found it had previously been applied in a mill in England as well as one in the United States. Also in 1898 he put in a reinforced-concrete floor for the Uncas Paper Company at Norwich, Conn., which he believed to be the first ever installed for a paper mill. One of the largest assignments of the firm was a paper mill with a 90-ton daily capacity, a 135-ton sulphite pulp mill, and a 70-ton soda pulp mill for the Oxford Paper Company at Rumford Falls, Me., this plant costing upward of one and a half million dollars.

In 1901 Mr. Wallace sold out his interest to Mr. Tower and opened an office in New York as mill architect and hydraulic engineer. The design and construction of a ground-wood pulp mill, with a daily capacity of 120 tons and a water power development of 15,000 horsepower for the Spanish River Pulp & Paper Co., in Ontario, gave him sufficient capital to form the firm of Joseph H. Wallace & Co. He built a number of mills in Canada, New Hampshire, New York, and Michigan in those early days of the company, and served with A. N. Burbank, treasurer of the International Paper Company, and Frank C. Trowbridge, president of The Black-Clawson Company, as a board of appraisers in connection with the consolidation of a large number of board paper mills, leading to the formation of the United Board & Paper Co.

At this time, also, he was participating in experiments in the manufacture of paper from cornstalks and sugar cane. With Dr. Viggo Drewson, an eminent chemist who had shown that the pith could be separated from the shell of cornstalks by a chemical process, and Joseph Warren, of S. D. Warren & Co., he conducted tests leading to the manufacture at the Warren mills at Cumberland, Me., of both book and coated papers. Papers made from Louisiana sugar cane exhibited at the Louisiana Exposition at St. Louis in 1904 by Wallace & Co. won a gold medal.

In 1903 Mr. Wallace was introduced to George Hitchen and Alexander McIvor, of the Wall Paper Manufacturers, Ltd., of Darwen, England, who were inspecting paper mills in the United States. He was asked to design improvements in the Darwen mill and went to England late in 1904. It was at Darwen that he found a paper machine which had been operating since 1870 with a twin engine of variable speed directly driving the back line, while another engine drove the pump line (at constant speed). He worked with Mr. Hitchen for a number of years, improving the Darwen mill and building a five-machine plant on the Thames River below London and operating it for three years.

During the years following 1904 Mr. Wallace traveled back and forth between the United States and Europe, building mills on both sides of the Atlantic. He established an office in London, with James Sickman in charge, to take care of his European enterprises. He visited France, Germany, Norway, and Sweden, finding much that was new to him and, in turn, contributing many improvements to the mills and to the manufacture of paper in those countries. He learned of the use of Fenestra, a steel product, instead of wood, for window frames and sashes in mills, bought the rights, and introduced the use of Fenestra in mill construction in the United States and Canada. At the North Wales Paper Company, at Flint, Wales, he was impressed by the vertical tower bleaching system, and later introduced it at the Greenhithe plant of the Wall Paper Manufacturers. This plant was entirely designed and equipped by Mr. Wallace and he operated it for more than a year after its completion.

At Moss, in Norway, Mr. Wallace saw sulphate mills in operation for the first time. He met Carl P. Carlson, who was the outstanding figure in the development of sulphate mills there, and secured his services as associate in the design of the first southern sulphate mill in the United States, for the Roanoke Rapids (N.C.) Paper Manufacturing Company. Mr. Wallace also built a paper mill and groundwood pulp mill, a power station and concrete dam, and a board mill for this and other paper companies in Roanoke Rapids.

In 1906, when Mr. Wallace was in Savannah, Ga., buying lumber for mills he was to construct, he noticed the burning piles of refuse from the sawmills. He became intensely interested in the utilization of this refuse. In articles by Mr. Wallace published in the *Paper Trade Journal* in 1931 (issues of January 29–April 16, inclusive) and later reissued by The Black-Clawson Company in its *Messenger*, he wrote extensively of his years of research in this field, out of which came the commercial production of kraft paper from all varieties of southern pine. He promoted the Southern Paper Company, designed and built its plant at Moss Point, Miss., and acted as vice-president and general manager of this mill for its first three years of operation.

Thoroughly convinced that the future of papermaking lay in the South, Mr. Wallace established laboratories near Stamford, Conn., and extended his research into the use of resinous pine wood for fine book and magazine papers. A miniature sulphate mill which was set up in the mill of the Southern Paper Company and run parallel to the big mill operations from 1913 to 1916, was moved to the Stamford laboratory in that year and was used during the World War in connection with tests for the Ordnance Department of the United States Army, with respect to fiber for explosives. Extensive experiments were conducted into the possibility of pine pulp from the South as a substitute for cotton linters and hull fiber.

In March, 1918, Mr. Wallace was appointed consulting engineer for explosives and was assigned to the design of Area E, at Nitro, W.Va., where the Government was to build a large smokeless-powder plant. The function of Area E was to receive raw material in the form of cotton linters and hull fiber and deliver it as bleached purified fiber to the Nitrating Department. Under the direction of Mr. Wallace three soda pulp mills, each with a daily capacity of 80 tons of bleached pulp, were completed by Armistice Day.

Mr. Wallace closed his London office early in the War, and from 1919 to 1925 gave much of his time to the development of processes for the profitable extraction of turpentine, rosin, and pine oil from southern pine in preparing a uniform raw material from these woods for the manufacture of high quality paper. His work in this direction represented a very important contribution to the industry.

From 1925 to 1930 Mr. Wallace was engaged in the manufacture of glassine paper, first as president of the Westfield River Paper Company, Russell, Mass., and subsequently as president of the Deerfield Glassine Company, Monroe Bridge, Mass. His work for The Black-Clawson Company since then related to the manufacture and sale of their complete line of paper and pulp mill machinery.

Mr. Wallace was the author of a book on "Pulp, Paper, and Power," published in 1909, and of many papers on pulp and paper manufacture. He also held numerous patents in this field. He became an associate-member of the A.S.M.E. in 1897 and a member four years later. He also belonged to the American Society of Civil Engineers, the Engineering Institute of Canada, and the Technical Association of the Pulp and Paper Industry, as well as to a number of other societies and organizations. Aside from his professional attainments he will be remembered for his broad interests, ranging from the fine points of cooking to boating, painting, and horticulture. He was a particularly joyous companion.

Mr. Wallace is survived by his widow, Mrs. Bernice (Read) Wallace, formerly of Petersburg, Ind., whom he married in 1927, and by his daughter by a previous marriage, Mrs. Jean (Wallace) Irving. His first marriage, in 1893, was to Clara H. Gunderson, of Worcester. He also left one sister, Miss Cora Wallace, of East Hartford, Conn.

EDWARD WESTON (1850–1936)

Dr. Edward Weston, inventor, scientist, and manufacturer of the electrical instruments bearing his name, died at his home in Montclair, N.J., on August 20, 1936.

Born at Brynn Castle, near Oswestry, Shropshire, England, on May 9, 1850, the son of Edward and Margaret (Jones) Weston, he was educated in the common schools of England and St. Peter's Collegiate Institute in Wolverhampton, and at an early age showed great interest in and aptitude for scientific matters.

His education along medical lines stimulated his interest in chemistry and his first experiments were in the electrometallurgical field.

As a young man recognizing the great opportunities in America, he left England and came to New York in May, 1870. After a short time spent in connection with chemical work, he became associated with the American Nickel Plating Company and later entered this business himself. His work in electroplating was of marked commercial significance and led him to a study of the sources of electrical energy available for plating, which interest led him naturally into a study of replacement of the primary battery by some other source of energy.

In a very short time he had made certain improvements in connection with the dynamoelectric machines and increased their efficiency to such an extent that he entered the field of manufacturing these machines, first for electroplating, and later for electric lighting. This factory was located in Newark, N.J., to which place he had removed in about the year 1875.

His study in connection with the dynamoelectric machine led to the securing of many important patents in connection with the construction of dynamoelectric machines, control devices, etc. His work in this field was characterized by a scientific approach and by care and precision unique in that day.

His study and manufacture of dynamoelectric machines directed his attention to the distribution and utilization of electrical energy for lighting, and he became a deep student of problems touching both incandescent and arc lights.

For a time he was engaged in the manufacture of various types of incandescent lamps and developed a technique which was of great importance in the advancement of the electric light industry.

Dr. Weston was not only a prolific inventor, securing more than three hundred United States patents, but he was an extremely clever and ingenious practical machinist, and was interested in all phases of research, invention, manufacture, and distribution. It was characteristic of his discoveries and inventions that they came very quickly into general use; Dr. Weston himself inventing the machines to manufacture his inventions and himself following through the problems of production and distribution.

All of the work that Dr. Weston did showed the necessity for electric measuring instruments of the highest precision, and commercial instruments of precision and rugged construction, and probably his greatest contribution to science and industry was the development of these electric measuring instruments and the Weston Standard Cell for which he is internationally famous.

The later years of his life were given over to independent scientific study and research and to the development of the electrical measuring instruments field, he being the largest producer of such instruments in the world. Wherever electricity is used, the name of Weston is known.

Dr. Weston was one of the founders of the American Institute of Electrical Engineers and the president of that institute in 1888. He had been a member of the A.S.M.E. since 1882, and was also a member of many other domestic and foreign technical societies, among which were the Electrochemical Society, the American Physical Society, the American Chemical Society, The Franklin Institute, and the National Electric Light Association (now the Edison Electric Institute).

He was, particularly in later years, very much interested in education, serving for many years on the Board of Trustees of the Stevens Institute of Technology and being the founder of the Newark Technical School which has lately developed into the Newark College of Engineering.

Dr. Weston was honored by the degree of LL.D. from McGill University in 1903, and the degree of Sc.D. from Stevens Institute of Technology in 1904 and from Princeton University in 1910.

He received many certificates and medals from various expositions in this and in foreign countries, and honors and awards in the electrical and allied fields came to him in great numbers. Many learned societies recognized his contribution to science.

He was recipient of the Elliott Cresson medal, awarded by The Franklin Institute in December, 1910, "for brilliant and successful research in the field of electrical discovery."

He was awarded the Perkin medal by the Society of Chemical Industry for his work in applied chemistry on January 22, 1915. He was recipient of the Franklin medal in 1924, and the 1932 Lamme medal of the American Institute of Electrical Engineers.

Dr. Weston was one of the first of the inventors to appreciate the importance of the scientific method of approach as relating to invention, and, perhaps more than any other individual inventor of this period, was actuated by the same general principles as characterize our modern research.

He was a man who cared nothing for advertisement, but whose contributions to science and industry were not only fundamental but far-reaching and long-lived.—[Memorial prepared by Allan R. Cullimore, Newark, N.J. Mem. A.S.M.E.]

WILLIAM RUSSELL FRANKLIN WHELAN (1876–1934)

William Russell Franklin Whelan, a member of the A.S.M.E. since 1915, died of heart disease in the Philadelphia (Pa.) General Hospital on September 8, 1934. Mr. Whelan was born in Boston, Mass., on August 9, 1876, and attended school there. After leaving high school he served a year's apprenticeship with the Braman Dow Company, Boston, and worked until 1898 on power plant installation and equip-

ment for the same company. From then until 1903 he was assistant operating engineer in the steam power and refrigerating departments of the Souther Brewing Company, Boston. He designed and installed a new system for the lubrication of the ammonia compressor piston rod of a De La Vergne "dry system" machine and designed and installed a vacuum system for heating water with exhaust steam.

From 1903 to 1912 Mr. Whelan was chief engineer for the estate of E. D. Jordan, Boston, in charge of all mechanical work in several plants operated by the estate. He designed and supervised the reorganization of the entire steam, hydraulic, and electric systems of one of these plants, including setting and piping of three 175-hp return tubular boilers.

Mr. Whelan left Boston in 1912 to become chief engineer for the International Paper Company, at Palmer, N.Y. He remained with the company for nearly ten years, his work including the design and installation of considerable new equipment and also of new systems, the making of exhaustive boiler trials and paper machine dryer tests, in order to improve practice, and other research work. For some years he was inspector of steam plants for the company.

The remainder of his professional experience was with the Crane Company, most of the time as engineer with the Philadelphia branch of the company.

THOMAS BYRD WHITTED (1876–1935)

Thomas Byrd Whitted, who died in New York, N.Y., on June 28, 1935, from a heart attack, was born in Rockingham County, North Carolina, on July 9, 1876, son of Thomas Byrd and Isabella (Scott) Whitted.

He received his early education at the Davis Military School, and completed it at the U.S. Military Academy, West Point. Resigning from the Army, he went with the Research Department of the General Electric Company in Schenectady, N.Y., in 1898, and two years later was transferred to the Denver office as district engineer. From 1902 to 1904 he was chief engineer of the U.S. Light & Traction Co., Denver. In 1904 he went with the Westinghouse Machine Company as manager of the office at St. Louis, Mo., remaining there until 1908, when he returned to his native state to engage in the sale of machinery, construction of steam and hydraulic plants, etc., establishing his office in Charlotte, N.C. He was still in this business at the time of his death, maintaining an office in New York, where he had spent most of his time during the latter part of his life.

He was widely known in machine and engineering circles, handling numerous projects in this country and in some instances abroad, including negotiations for the sale of the hydro-equipment to the Russian Government for the Dnieper River installation, which was recently put in service.

Colonel Whitted never entirely lost his contact with his first love, and throughout his life maintained a great deal of interest in all military matters. After leaving West Point he was, while a resident of the State of New York, an officer of the New York National Guard. Later, while in the West, he was a member of the Colorado National Guard, and on his return to North Carolina in 1908 he was engineer officer of the North Carolina National Guard for some years, and later colonel acting on Governor Craig's staff.

He was commissioned in the Officers Reserve Corps, U.S. Army, in June, 1917. He applied for duty on declaration of war, and was assigned to active duty on May 8, 1917. He was ordered to France in the summer of 1917 as captain in the Engineer Section, Office of Chief Engineer, A.E.F., and assigned to the staff of General Dawes, in charge of purchases and supplies. He was later promoted to major, and was assigned to active duty on the front with French troops. He was instructed to return to this country with other officers to train combatant troops for the Engineer Division. He was promoted to lieutenant-colonel, and was training troops at Fort Humphreys when the Armistice was signed. He retired from active service on November 29, 1918, and remained until his death a major in the Engineer Section.

Colonel Whitted received through the War Department of the U.S. Army notice of his being cited by the French government and decorated for the excellence of his work as an officer of the Corps of Engineers, U.S. Army, in France during the World War. The citation certificate is of the Order of University Palms, carrying with it the decoration of Silver Palms, "Officier d'academie" and silver palms.

Colonal Whitted is survived by his widow, the former Annie Cornelia Tate, of Charlotte, N.C., whom he married in 1901, a son, Captain Thomas Byrd Whitted, Jr., of the U.S. Army, a daughter, Margaret Overman Whitted (Mrs. T. E. Efird), of Charlotte, N.C., his mother, and also a brother, Commander W. S. Whitted, U.S.N.(Retired).

Colonal Whitted became a junior member of the A.S.M.E. in 1900, and associate-member in 1903, and a member in 1909. He was also a member of the American Society of Military Engineers and the Veterans of the World War. He was a Mason and a Shriner; a member of the Manufacturers Club and the Charlotte Country Club of Charlotte, N.C.; and of the Engineers' Club, the Bankers Club, and the Wingedfoot Country Club of New York.—[Memorial prepared by WENDEL W. CLINEDINST, New York, N.Y. Mem. A.S.M.E.]

FRANCIS P. WILSON (1868–1932)

Francis P. Wilson, chief operating engineer of the Illinois Maintenance Company, Chicago, Ill., died on September 5, 1932.

Mr. Wilson was born on July 19, 1868, at London, Ontario, Canada. He supplemented his early education with courses through the International Correspondence Schools. Prior to his association with the Illinois Maintenance Company he spent a year each with the Canadian Pacific Railroad Company and the Chicago Water Works, in construction work, five years as a marine engineer on inland waters, holding a chief engineer's license, and ten years as assistant operating engineer for the Union Traction Company.

Mr. Wilson became assistant chief engineer of the Illinois Maintenance Company in 1908 and chief operating engineer in 1918. He had complete charge of the operation of a large number of steam generating plants which form the central station steam heating and power system of the Illinois Maintenance Company, a subsidiary of the Commonwealth Edison Company.

Mr. Wilson became a member of the A.S.M.E. in 1923.

JULIAN ERNEST WOODWELL (1874–1935)

Julian Ernest Woodwell, who had been engaged in the practice of consulting engineering at New York, N.Y., for 27 years at the time of his death in that city on October 23, 1935, was a native of Maine. He was born at Wells on January 7, 1874, and prepared for college at the high schools of Sandwich and Newburyport, Mass. In 1896, he was graduated from the mechanical engineering course at the Massachusetts Institute of Technology with the degree of S.B.

Mr. Woodwell was resident engineer at Washington, D.C., for S. H. Woodbridge, a consulting engineer of Boston, Mass., and supervised improvements in the heating and ventilating system of the Capitol in 1896 and 1897. The following year was spent in office work relating to the design of boiler plants and heating and ventilating systems which were installed in schools and other buildings in the vicinity of Boston.

Early in 1898, Mr. Woodwell left the employ of Mr. Woodbridge to accept a position in the U.S. Treasury Department, where he had jurisdiction over the operation of all Federal buildings throughout the United States for slightly more than a year. In that capacity, he designed isolated electric light and power plants for the post offices at Baltimore, Md., Pittsburgh, Pa., and Louisville, Ky., and prepared plans and specifications for their installation and the complete rewiring of these buildings for electricity. From 1899 to 1908, Mr. Woodwell was inspector of electric light plants in all public buildings under the control of the Treasury Department, his duties corresponding to those of a chief engineer in the purchase of fuel, light, water, and supplies and the control of between 400 an 500 operating engineers and their assistants. In this ten-year period, the approximate number of buildings under his supervision increased from 400 to 1000. In this connection, he designed complete electric light and power plants for the Federal Building at Philadelphia, Pa., the post offices at New Orleans, La., St. Louis, Mo., and Cincinnati, Ohio, and the Appraisers' Stores in New York, N.Y., and supervised the installations as well as complete or partial rewiring of these buildings. He also designed additions to the power plants in the post offices at New York and Boston, which included new units and switchboards, and prepared plans and specifications for a complete rewiring of the Treasury Department Building, Washington, D.C. All of these installations were made under his direction. While at Washington, Mr. Woodwell also designed and supervised the installation of a complete central heating, lighting, and power plant to serve the Patent Office, Pension, and General Land Office Buildings; and designed and prepared plans and specifications for an electric light and power plant and the lighting system for the National Museum, one unusual feature of this last being the use of lead-covered cable installed in a tile duct adjacent to the footing walls and in accessible tunnels for horizontal runs of the principal feeders to the various distribution panels. In 1905 and 1906, he was associate consulting engineer for the Onondaga Country Court House, Syracuse, N.Y., and designed the hydraulic elevator plant, a 300-kw electric light and power plant, and the electric wiring and fixture equipment of the building.

In June, 1908, Mr. Woodwell resigned his Government position and formed a partnership with L. B. Marks to practice consulting

engineering in New York. The firm was commissioned by McKim, Mead & White to prepare plans and specifications for and supervise the installation of the complete mechanical and electrical equipment, including forced hot-water heating system, ventilating equipment, electric power and light wiring, elevator equipment, and clock, telephone, fire-alarm, and illumination systems, for the new post office at New York. The execution of this commission required four years (1909–1913). In connection with the work at the post office, Mr. Woodwell designed, prepared plans and specifications for, and supervised the installation of a complete mail-handling system between the post office and the platforms in the adjacent station of the Pennsylvania Railroad. This system, which was the first of its kind in the United States, uses spiral chutes, parabolic slides, and horizontal transfer belts to bring mail in large tonnages from the floor of the post office directly to the mail-car doors. Incoming mail is handled in a specially designed bucket lift. Between 1911 and 1914, Mr. Woodwell was in charge of the installation of wiring systems for the Congressional Library and the State, War, and Navy Building in Washington, and the complete mechanical and electrical equipment for the Army and Navy and University clubs of Washington and the Chemists' Club, New York.

In 1915, Mr. Woodwell withdrew from the partnership with Mr. Marks and engaged in independent practice. This work included the design of forced hot-water heating systems, building construction, power plant design, and a 12-year engagement as consulting engineer for the city of Lansing, Mich., in the development and construction of municipal steam-heating and electric light and power projects. Mr. Woodwell was retained by the State of Pennsylvania at various intervals from 1914 to 1932 in connection with engineering work for the State Hospitals at Warren, Polk, Harrisburg, and Allentown. During this same period he was also retained by Sidney Blumenthal, Inc., in connection with their several industries.

In addition to the A.S.M.E., of which he became a junior member in 1900 and a member in 1908, Mr. Woodwell belonged to the American Institute of Electrical Engineers, Illuminating Engineering Society, American Society for Testing Materials, and Michigan Engineering Society.

Mr. Woodwell is survived by his widow and two sons.

Operating Experiences in the Steam and Power Department of the South's First Alkali Plant

By G. R. AVERY,[1] CORPUS CHRISTI, TEXAS

In this paper is given a simplified diagrammatic outline showing the water and steam flows in the powerhouse of the Southern Alkali Corporation. The greater part of the boiler feedwater is raw river water carefully treated by the hot-process lime-and-soda-ash method followed by a phosphate treatment directly into the boiler drum. Excellent results have been obtained with this type of treatment as is shown by the present maximum length of run allowed before the boiler is taken out of service for inspection as a matter of safety procedure. Costs due to boiler outage, maintenance of boiler and furnace, labor and material have been considerably reduced by this close attention to feedwater treatment and control.

Electric power generation is obtained as the by-product of steam used in process requirements. Turbine blading has shown signs of corrosion due to moisture and probably will be replaced with stainless steel or its equivalent when necessary. The protection of generators and motors when not in use is a serious problem along a moist and humid coast. They are protected in this instance by covers and electric heaters. Pump and air-compressor valve troubles have been mainly a question of materials and seem to be fairly satisfactorily settled now.

Automatic control of boiler and water pressures is used extensively with good results. Automatic alarms guard against the contamination of condensates being returned from process departments. Excellent cooperation in the design, maintenance, and operation of the plant have resulted in a smooth load curve with high load and use factors.

THE FIRST alkali plant of the South went into production during the latter part of 1934 and the usual difficulties consequent to starting a new plant were encountered. Certain adjustments and changes were necessary before all the apparatus operated satisfactorily.

Equipment and Design

The steam and power plant as designed includes two 1200-hp boilers at 400 lb per sq in. gage, 60 F total temperature; two 2500-kw turbines; two main boiler feedwater pumps; two feedwater booster pumps; two plant fresh-water pumps; water treatment and proportioning equipment, together with other miscellaneous equipment.

There is no building partition between the boilerhouse and turbine room because the use of natural-gas fuel allows a high degree of cleanliness.

The boilers are of the X-drum type with sectional headers and above-deck superheaters, having air-cooled sides and front walls with a forced-draft fan blowing air under the furnace floor to the wind box on the front of the boiler. No economizers or air preheaters are used at present.

General Boiler and Furnace Data

Boiler maker: Babcock and Wilcox Company.

Type of boiler: Cross-drum, water-tube, sectional headers above-deck superheaters; and B. & W. steam scrubbers in drum.

Steam and water drum: 60 in. diameter, 22 ft long, one per boiler.

Boiler tubes: 4 in. diameter, 30 tubes wide, 14 tubes high.

Superheater tubes: 2 in. diameter, four-loop, three-pass.

Heating surfaces: Boiler, 11,756 sq ft; superheater, 4000 sq ft.

Furnace data: Width 17 ft, 10 in.; volume 7860 cu ft.

The main generator and feedwater-pump turbines are 400 lb gage admission and 15 lb gage exhaust with a bleeder at 150 lb gage on the main generator turbine.

Steam at 150 lb gage and 15 lb gage is supplied to the main plant headers feeding to the plant process departments.

The flow diagram, Fig. 1, shows the water and steam distribution in the powerhouse.

Water flows from the plant's earthen reservoir to the fresh-water pumps. Here it is pumped to the process departments and to the sedimentation tank. In the sedimentation tank it is treated by the hot-process lime-and-soda-ash method and also heated to 230 F. The feedwater booster pump takes the treated hot water from the sedimentation tank, pumps it through the filters and into the deaerator tank. It is heated further to 250 F and then goes to the main feedwater pump where it is pumped to the boilers.

The feedwater pressure required to feed the boilers is maintained by changing the speed of the turbine driving the pump, thereby increasing or decreasing the pump-discharge pressure developed. This is accomplished automatically by a constant excess-pressure, diaphragm-operated regulator which opens or closes the steam valve in the line to the turbine.

A small high-pressure pump feeds di-sodium phosphate solution directly to the boilers. This solution is made by adding from 25 to 35 lb of anhydrous di-sodium phosphate to 100 gallons of water. A mercoid switch control operates intermittently

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to maintain the phosphate solution at the same level as that of the chemical solution in the proportioner supply tank, thus obtaining a constant ratio between the volumes of chemical solution and phosphate solution added per gallon of make-up water. The chemical solution consists of soda ash and lime and sodium sulphate when required. There is a 4 to 6-in. lag in solution levels between pumping intervals according to the adjustment on the spring which controls the trip on the mercoid switch.

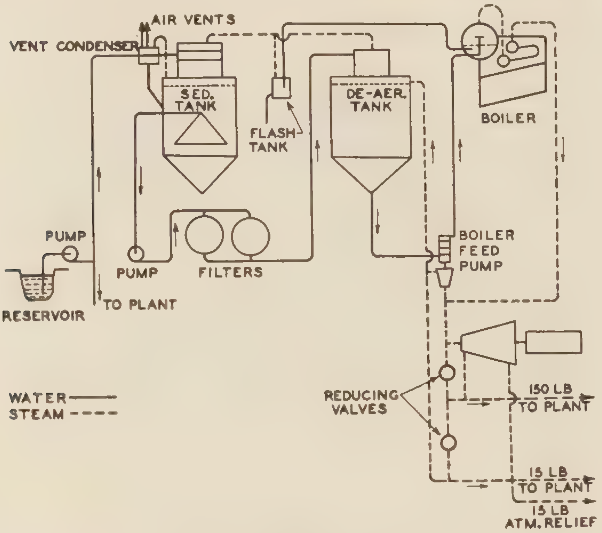

Fig. 1 Flow Diagram Showing Water and Steam Distribution in Powerhouse

The charge of phosphate is usually adjusted to give an excess of 50–100 ppm of phosphate in the boiler water.

Operation

Feedwater Treatment. Based on the hypothesis that good external water treatment will keep boilers free from trouble, the water tests shown in Table 1 are made daily at the following times:

Raw and treated-water test, every 2 hours
Boiler-water test, every 3 hours
Boiler-water total solids test, every hour (by hydrometer)

Fig. 2 shows a yearly record of these tests. The points plotted are the weekly averages of the daily tests shown in Table 1.

TABLE 1 WATER TESTS

Test and samples	Method of analysis	Expression of results
Hardness		
Raw water......	Soap solution	Grains per gal (as $CaCO_3$)
Treated water...	58.3 ml sample 1 ml = gr per gal.	
Methyl-orange alkalinity		
Raw water......	N/29.15 HCl	Grains per gal (as $CaCO_3$)
Treated water...	100 ml sample methyl-orange indicator	
Chlorides		
Raw water......	0.01715 N $AgNO_3$	Grains per gal (Cl as NaCl)
Deaerated water	58.3 ml sample	
Boiler water.....	K_2CrO_4 indicator	
Phenolphthalein alkalinity		
Treated water...	N/29.15 HCl	Grains per gal (as $CaCO_3$)
Boiler water.....	100 ml sample phenolphthalein indicator (boiler water) in presence of excess $BaCl_2$	
Boiler water		
Total solids...	By hydrometer	Grains per gal
Per cent blowdown	Ratio of chlorides	Per cent

To check the effect of returning blowdown to the system, bimonthly analyses, shown in Table 2, are made on raw water, treated water, and boiler-water samples for the following constituents:

(1) Insoluble silica, as ppm SiO_2 tested gravimetrically
(2) Al_2O_3, as ppm Al_2O_3 tested gravimetrically
(3) Soluble silica, as ppm SiO_2 tested gravimetrically
(4) pH of boiler water and steam, tested by electrometric potentiometer

Condensate returns from the caustic-soda department are continuously checked for caustic contamination by a conductivity meter. The purpose of this precautionary measure, which has given excellent results, is to guard against high boiler-water alkalinities.

Fig. 3 is a record of the chemicals used per thousand gallons of make-up water; from this a cost figure per thousand gallons can be computed, based on the existing market prices of the chemicals.

TABLE 2 WATER ANALYSES AND pH VALUES

Date	Dissolved SiO_2 ppm		Insoluble SiO_2 ppm		Al_2O_3 ppm		pH	
	Raw	Treated	Raw	Treated	Raw	Treated	Blr Water	Blr Steam
9-5-35	12.3	14.7	9.8	..	0.7	0.9	..	..
9-20-35	15.8	11.0	25.9	4.0	12.4	2.1	..	..
10-7-35	14.9	23.1	30.1	0.9	8.2	0.9	..	..
10-22-35[a]	20.0	14.3	7.1	2.4	0.6	0.1	..	..
11-5-35[a]	20.7	13.4	1.4	0.0	0.8	0.8	..	..
11-20-35[a]	17.8	7.8	5.3	0.0	0.6	0.1	..	..
12-5-35[a]	17.9	12.6	4.6	0.0	6.5	1.2	..	..
12-20-35	23.1	14.8	0.0	0.0	2.4	1.3	..	..
1-6-36	15.7	14.1	6.7	0.0	3.4	2.8	..	..
1-20-36	15.2	7.5	10.9	0.0	1.6	1.2	..	..
2-5-36	20.8	7.4	7.3	0.0	1.1	..	..	..
2-20-36	10.1	5.1	9.8	0.0	4.4	3.2	..	..
3-5-36	11.1	9.0	46.1	0.0	0.8	1.2	..	..
3-20-36	10.7	5.4	12.7	0.0	1.3	0.9	..	..
4-6-36	10.9	4.9	73.4	0.0	0.6	0.3	..	..
4-20-36	10.9	2.9	4.0	0.0	1.8	1.7	..	..
5-5-36	11.8	3.2	9.1	0.0	1.0	1.4	..	..
5-21-36	13.1	5.2	6.4	0.0	0.5	0.7	9.5	7.3
6-5-36[a]	20.3	15.3	40.4	0.0	0.6	0.3	9.5	5.7
6-22-36[a]	21.5	15.7	8.2	0.0	0.8	0.7	9.5	6.2
7-3-36[a]	26.8	21.1	8.4	0.0	3.8	2.1	9.3	4.6
7-20-36[a]	24.8	26.7	17.5	0.0	2.6	1.7	9.1	5.6
8-21-36[a]	36.1	24.9	6.0	0.0	2.9	1.9	9.4	6.1
9-5-36[a]	35.8	21.5	7.0	0.0	2.6	2.5	9.6	..
9-21-36[a]	28.3	19.3	10.4	0.0	2.7	1.1	10.1	6.7
10-5-36[a]	26.9	21.3	53.0	0.0	1.6	1.1	9.3	6.0
10-20-36[a]	21.6	16.4	22.8	0.0	3.9	2.3	10.0	6.8
11-6-36[a]	20.6	14.9	11.1	0.0	1.3	1.3	9.4	7.3
11-20-36[a]	38.2	19.2	7.2	0.0	1.4	1.2	9.2	5.1

[a] Continuous blowdown being returned to raw water.

TABLE 3 BOILER SLUDGE AND SCALE ANALYSES

Boiler-Sludge Analysis, August 20, 1934
No. 2 Boiler

	Per cent
$Ca_3(PO_4)_2$	41.5[a]
$AlPO_4$	28.1
$CaCO_3$	4.6
$CaSO_4$	2.8
SiO_2	1.7
Fe	9.6
Al_2O_3	11.7
	100.0

[a] By weight.

Boiler-Scale Analysis, August 16, 1934
Top row of boiler tubes, about 2 feet from circular headers on upper part of tube

	Per cent
SiO_2	2.26
R_2O_3	91.75[b]
$CaCO_3$	4.12
$MgCO_3$	1.68
$CaSO_4$	4.40
	104.21

[b] Includes metallic iron as well as oxide of iron and aluminum.

The true value of water conditioning is, of course, in the condition of the boiler and boiler tubes. Fig. 4 shows the periods of operation between boiler-tube cleaning intervals. The present period of operation is approximately six months. As a safety measure, the boilers are cleaned more at these times than is justified by the amount of scale found on the tubes. Nor-

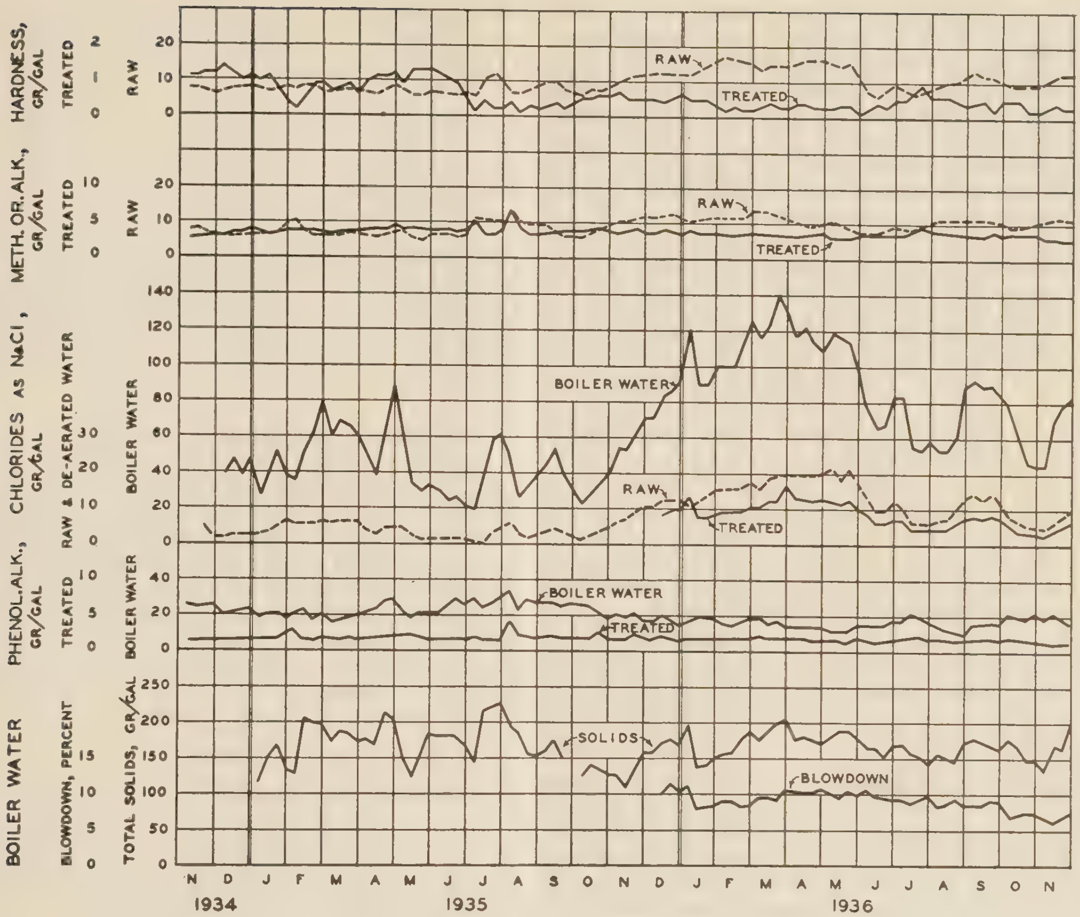

FIG. 2 ANNUAL RECORD OF WATER TESTS

mal operation consists of having in service at all times both of the boilers. They are cleaned and inspected at the stated intervals, shown by Fig. 4, one immediately after the other.

Rigorous inspection with continuous observation of the condition of the boilers is felt justified from the point of stand-by equipment.

During the initial period of operation and before the treated water was clear, there was a soft sludge and a hard scale depositing on the tubes. A typical analysis of each of these materials is shown in Table 3.

It will be noted that Fig. 4 shows the periods of service before June, 1935, as 35–50 days. During this time great difficulty was experienced in obtaining a clear treated water. The raw water coming from the Nueces River source showed a high turbidity and a fairly low hardness and chloride content. By adding extra coagulating materials to the sedimentation tank it was possible to improve the clarity of the water. The coagulants tried were aluminum sulphate, sodium aluminate, and magnesium sulphate. However, there still remained a considerable turbidity in the treated water until the hardness and chloride content of the raw water increased. Immediately the treated water became clear. Several opinions have been ad-

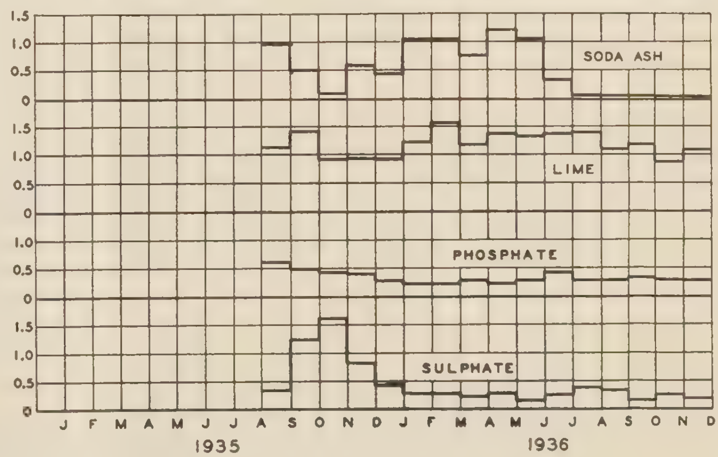

FIG. 3 CHEMICALS USED PER THOUSAND GALLONS OF MAKE-UP WATER

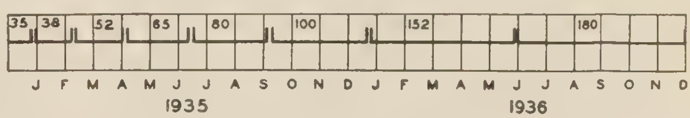

FIG. 4 LENGTH OF BOILER RUNS

vanced to account for this, one of which was that the low-hardness, low-chloride water did not form a sufficient quantity of floc to remove the very fine particles with it in settling. Another was that the increase of total solids aided in forming a coarser precipitate or floc; as a result better clarifying action was obtained as the floc settled.

Whatever the cause, the use of extra coagulants has been discontinued. During periods of low chloride concentration in the raw water we return the boiler continuous blowdown to the water reservoir which builds up the amounts of chloride and solids in the raw water. To the present time we have had no more trouble with turbidity in the treated water.

As a check on the rate of settling in the sedimentation tank and capacity per sq ft of filter area, the following normal and maxium operating rates are shown:

	Normal	Maximum
Settling time, sedimentation tank....	3 hr, 10 min	1 hr, 10 min
Water per sq ft of filter area........	43.7 gal	83.5 gal

It seems evident from this that operating the softener at too high a rate should not have been a factor in preventing the clearing of the turbid water.

Boiler output ranges from 175 per cent of normal rating to 325 per cent maximum. These ratings are not excessive and all conditions favor the production of clean steam.

Since the difficulty with turbid treated water has been overcome the length of runs has been increased as shown in Fig. 4, slowly enough to enable checking the condition of the boiler tubes.

The steam scrubbers are apparently working satisfactorily, although the boiler ratings have not been high enough for a more thorough check.

Heat recovery of the continuous blowdown may be installed later. At present this is flashed to the steam space in the sedimentation tank and the remaining liquid is returned to the sewer or to the water reservoir, depending on the raw-water conditions.

To obtain a uniformly treated boiler feedwater in the face of rapid and extreme fluctuations in the raw water calls for frequent testing with similarly frequent changes in the amounts of chemicals fed to the treatment tank.

The continuous blowdown from the boiler is manually regulated, depending either on the boiler-water alkalinity or on the total solids tests; this choice is governed by the load carried on the boilers.

It may be noted that the per cent blowdown has a rather high value, but it seems justified by boiler-water and tube conditions. This value has been varied for several trial boiler runs. However, the existing condition appears economically justified when the present low costs of water and fuel are considered together with the restricted stand-by equipment and the isolation of the plant from a repair-parts point of view. Oxygen corrosion is not at all apparent on the boiler drums or tubing.

Turbines

The amount of steam carry-over is negligible. Deposits on the turbine blading at present are not noticeable. However, during the first year when boiler-water alkalinities were high, there occurred a considerable deposit which required hand scraping and brushing. This scale had an analysis as shown in Table 4.

The carry-over of solids occurred while there was a turbid treated water and high alkalinities were being maintained in an attempt to prevent sodium aluminum silicate from depositing on the boiler tubes.

There has been trouble with the rusting and pitting of the turbine blading, due possibly to steam leakage through shut-off valves, steam traps, or to atmospheric moisture. Since the present blading is not stainless steel, it is proposed to replace it

TABLE 4 TURBINE SCALE SAMPLE ANALYSIS (BY WEIGHT)

	Per cent
$NaCl$	61.10
Na_2SO_4	19.08
$NaOH$	0.40
Na_2CO_3	8.79[a]
$CaSO_4$	5.41
$MgSO_4$	0.61
SiO_2	3.49
R_2O_3	0.58
Moisture	0.54

[a] By difference.

with stainless-steel blading, or its equivalent, whenever it becomes necessary. Vent openings have been arranged between double valves on the pressure lines to the turbine. These are to remove the possibility of steam leakage into the turbine. The turbine traps have been altered so that they discharge to the atmosphere.

Generators and Motors

To keep the generator and motor windings dry electric heaters are installed in the generators and in the motor bases. The generator heater is mechanically interlocked with the exciter switch so that when the exciter switch is closed the heater is turned off, and when open it is turned on. The results of the Megger tests indicate that this accomplished the desired results. Before installing the heaters, a Megger reading showed 150,000 ohms; after the installation, the reading was increased to one to two million ohms. When not in use, the motors and exciters are protected by canvas covers and electric heaters in the base to keep out the moisture. Along the coast, the moisture and high humidity provide ever-present problems.

Pumps

No trouble has been experienced with the fresh-water pumps, but it has become necessary to use an acid-resistant bronze for sea-water pumping purposes. There are times during the year when a small amount of sulphide is present in the sea water. Aside from normal wear, little further trouble has occurred since the installation of the bronze equipment.

The boiler-feed-pump shaft sleeves, originally of bronze, broke seemingly because of crystallization. This may have been caused by heat generated by the friction of the packing on the sleeve. They were replaced with new monel-metal sleeves which are standing up well. A nitrided steel would probably be as good or better.

Since there are only two boiler feed pumps, a close check is kept on their condition, and repairs are made immediately as needed. This repair work is always continuous until the job has been completed. No trouble is anticipated if the maintenance work is kept up to the minute. With the exception of the previously mentioned difficulties, the main feed pumps, pumping water at 250 F and 460 lb per sq in. gage, have given little trouble. This may be due in part to the pressure developed per stage, amounting to 100 to 125 lb. A small amount of water is bled from the second stage for process use.

Air Compressor

The air compressor is a constant-speed, motor-driven, two-cylinder, double-acting unit with an intercooler and a three-step unloader. Best results have been obtained with stainless-steel valves.

Control

Feedwater-treatment control is accomplished with a proportioner-type controller. Changes in the water condition are met by changing the weights of chemicals used per charge.

The water pressure to the sedimentation tank is controlled by pilot air-operated control valves.

Boiler control is of the air-operated type, with the air normally being furnished by the plant air system. An emergency steam-driven air compressor will automatically start in case of power failure or low air pressure.

The firing equipment consists of ring-type combination oil and gas burners, operating at present on natural-gas fuel.

Fuel and air flow are controlled by air pressure from the master pressure controller and by a separate correcting adjustment for increasing or decreasing the air control to the uptake damper. This separate correcting adjustment is made from a relay attached to the steam-flow-air-flow ratio beam in the boiler meter.

Daily instrument charts representative of normal boiler operation show that for variations in steam flow between 45,000 and 65,000 lb per hr the steam pressure is held within a 5-lb range, and that during a sudden drop of 5000 to 10,000 lb per hr in steam flow the steam pressure will increase only 3 lb per sq in.

All gages and control units are centralized on the panel board which tells the complete story, thus instantly locating the source of any irregularity.

The main difficulties experienced with control have been in stopping the air leaks in the system and in adjusting the controls to the same lag or speed of response as the equipment being controlled. This has been fairly well completed, although small adjustments are normally necessary depending on the weather and the number of burners in service.

In case of a power failure, direct-current, solenoid-operated valves function in the control line to open the induced-draft damper and to close off the fuel-regulating valve to a predetermined point. The load is then carried by manual operation until automatic operation is restored and the fan motor is again in operation.

Close cooperation between the departments which utilize steam and the boilerhouse makes for a steady load with no excessive peaks or valleys in the steam demand.

Summary

Due to good engineering judgment exercised in designing the plant, equipment, and the layout, and to the observance of a strict maintenance program, it has been possible fully to realize the performance expected of the original equipment.

Careful control, testing, and analysis of boiler waters and the continuous blowdown control to hold the water at the condition desired has resulted in virtually eliminating the turbining of boiler tubes.

Acknowledgment

The author wishes to acknowledge his indebtedness to G. Heinemann and to H. Rahn of the Southern Alkali Corporation for their comments and aid on the chemical terminology, and to R. S. Clark of the Pittsburgh Plate Glass Company of Barberton, Ohio, for his helpful assistance.

A New Low-Alloy Pearlitic Steel for High-Temperature Service

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This paper describes the high-temperature characteristics of a carbon-molybdenum steel which differs from those now in general use in that it contains 1.5 per cent rather than from 0.5 to 1 per cent molybdenum. On the basis of standard creep tests and comparative constant-loading tests, it is shown that the increase in molybdenum content has greatly improved the load-carrying ability of the steel, especially at the higher temperatures.

Since the oxidation and corrosion resistance of this steel is no better than that of plain carbon steels, advantage cannot be taken of its improved load-carrying ability unless its surface is protected from attack. Accordingly, this analysis is considered only in a calorized condition. The present-day calorizing process is briefly described and certain proposed applications for this calorized steel are discussed.

THE PAST few years have seen considerable progress made in the development of low-alloy pearlitic-type steels for high-temperature service. Through a proper selection of analyses now available, suitable combinations of creep strength, oxidation and corrosion resistance can be obtained for many applications in which formerly only high-alloy ferritic or austenitic steels were considered. While certain of these steels are rendering satisfactory service there is, of course, an ever-existing need to obtain steels with even better load-carrying ability combined with a higher degree of surface stability. Only in this way can the use of the steels of this type be further extended to higher operating temperatures or to more severe conditions under present operating temperatures.

The present paper considers the high-temperature characteristics of a 1.5 per cent molybdenum steel in the calorized condition. The molybdenum, in this amount, provides the strength characteristics while the calorized surface produces a high degree of oxidation and sulphur-corrosion resistance. In the uncalorized state this analysis would possess an oxidation resistance no better than that of a plain carbon steel and in this condition its use would be confined to a maximum operating temperature of 950 to 1000 F. When the surfaces are calorized, however, this analysis can be used under properly designed stresses for prolonged service at temperatures up to 1200 to 1300 F.

Material Considered

The steel used in this investigation was made in a commercial electric-arc furnace in a charge of 2000 lb. It was cast into 100-lb ingots and hot-rolled into $^7/_8$-in. bars. The steel was composed chemically of 0.10 per cent carbon, 0.28 per cent manganese, 0.12 per cent silicon, 0.017 per cent phosphorus, 0.016 per cent sulphur and 1.38 per cent molybdenum.

The necessary creep, tensile, and impact specimens were then calorized, commercial equipment again being used. The calorizing process consists of a series of operations, and since these have undergone pronounced changes during the past three years, they will be briefly described.

Calorizing Process. The modern calorizing process consists of two operations. In the first, the parts to be calorized are heated in closed retorts with the correct calorizing mixture and an iron-aluminum alloy is formed on the surfaces. This contains from 50 to 75 per cent aluminum but has a depth of only 0.005 to 0.010 in. Furthermore, this layer is brittle and a distinct line of demarcation exists between it and the base steel. Therefore, it will spall off readily when subjected to deformation.

In the second operation, the calorized parts are maintained at a proper temperature for a sufficient period of time to cause the alloy initially formed to further diffuse into the base steel. The degree of diffusion can be controlled and the conditions are generally adjusted so that the aluminum concentration at the surface is reduced to 30 per cent, and the depth of diffusion increased to 0.035 in. Thus, this second operation greatly improves the ductility of the surface alloy by decreasing the aluminum concentration and by eliminating the line of demarcation between the base steel and the surface alloy.

The two heating periods required in the calorizing operations will, in general, coarsen the grain structure of the base steel. After the completion of calorizing, it is necessary to refine the coarsened structure by annealing.

Experimental Results

Even though a steel may possess a good combination of high-temperature characteristics it must also have suitable room-temperature properties in order to permit fabrication and installation. Therefore, it is necessary to consider both the room-temperature and high-temperature properties.

Room-Temperature Properties. Table 1 gives the room-temperature tensile and impact properties of the 1.5 molybdenum steel in both the calorized and uncalorized condition. First, considering the uncalorized material, it will be noted that a suitable combination of strength and ductility exists in both the as-received (hot-rolled) and annealed conditions. Even though the

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[2] Vice-President, The Calorizing Company, Mem. A.S.M.E. Mr. Brown was graduated from Cornell University with an M.E. degree and then served three years as production engineer with the Hyatt Roller Bearing Company, Newark, N. J. During the war he served in the Ordnance Department of the Army as a first lieutenant and was stationed at the Frankford Arsenal as production engineer in the manufacture of fuses and other small munitions. After the war, he spent five years at the Barlow Foundry, Newark, N. J., as engineer and assistant to the general manager. Since 1924, he has been associated with the Calorizing Company and served as engineer and district manager before taking his present position.

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Discussion of this paper should be addressed to the Secretary A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1937, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TABLE 1 INFLUENCE OF HEAT-TREATMENT ON THE ROOM-TEMPERATURE PHYSICAL PROPERTIES OF UNCALORIZED AND CALORIZED 1.5 MOLYBDENUM STEEL

Condition of specimens	Tensile strength, lb per sq in.	Yield point, lb per sq in.	Yield stress, lb per sq in., 0.2 %	Proportional limit lb per sq in.	Elongation in 2 in., per cent	Reduction of area, per cent	Charpy impact, ft-lb
Uncalorized:							
As received:							
No. 1	75500	47500	43500	28500	31.50	69.30	113.0
No. 2	75250	45000	43000	26600	34.00	69.70	104.0
No. 3	...	...	...	...	...	...	108.0
Average	75375	46250	43250	27550	32.75	69.50	108.3
Annealed at 1650 F and drawn at 1300 F:							
No. 1	64400	44900	43000	31000	37.00	73.30	142.0
No. 2	64600	45000	43000	32500	36.00	73.00	145.0
No. 3	...	...	...	...	...	...	147.0
Average	64500	44950	43000	31750	36.50	73.15	144.7
Calorized:							
As calorized:							
No. 1	58700	33600	31200	7100	26.00	28.10	...
No. 2	58700	30700	28800	7100	24.00	37.00	...
Average	58700	32150	30000	7100	25.00	32.55	...
Annealed at 1650 F:							
No. 1	58000	30700	28800	7100	25.50	33.70	...
No. 2	58000	33000	28800	7100	29.50	41.10	...
Average	58000	31850	28800	7100	27.50	37.40	...
Annealed at 1650 F and drawn at 1300 F:							
No. 1	57400	31200	30400	16900	27.00	45.50	52.0
No. 2	57100	31200	28900	16900	27.00	46.90	60.0
No. 3	...	...	...	...	...	...	60.0
Average	57250	31200	29650	16900	27.00	46.20	57.3
Calorized, but calorizing removed:							
Annealed at 1650 F and drawn at 1300 F:							
No. 1	63500	37350	...	...	31.50	71.20	...
No. 2	63600	37450	...	...	32.00	73.50	...
Average	63550	37400	...	...	31.75	72.35	...

NOTE: All values for calorized specimens are based on total cross-sectional area; that is, metal plus calorized layers.

annealing teatment spheroidized the structure, the tensile strength is greater than 60,000 lb per sq in. and the yield stress in excess of 40,000 lb per sq in. The ductility is likewise good, the elongation being greater than 35 per cent and the reduction of area in excess of 70 per cent.

The Charpy impact resistance of this steel is likewise outstanding. In the as-received condition it is 108.3 ft-lb while the annealing treatment increased this value to 144.7 ft-lb.

TABLE 2 BRINELL HARDNESS OF UNCALORIZED CYLINDERS WATER-QUENCHED AND AIR-COOLED AFTER HEATING TO TEMPERATURES BETWEEN 1450 AND 1750 F

Heating temperature, F	Brinell hardness after designated rate of cooling: Water-quenched	Air-cooled
As received	...	150
1450	163	137
1550	240	146
1650	293	146
1750	297	153

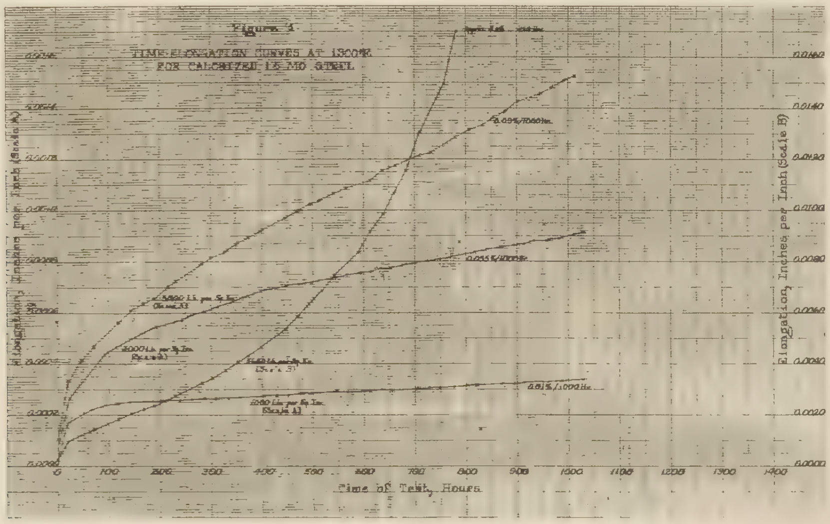

FIG. 1 TIME-ELONGATION CURVES AT 1300 F FOR CALORIZED 1.5 MOLYBDENUM STEEL

In the majority of cases, the calorizing treatment lowered the strength, ductility, and impact characteristics. Even in the calorized condition, however, this 1.5 molybdenum steel possessed an elongation of 25 per cent. The annealing treatment raised the elongation of the calorized specimens only slightly, but did exert a marked influence on the reduction of area, raising this value from 32.55 to 46.2 per cent. The impact resistance, while considerably below that of the uncalorized steel, is good, being 57.3 ft-lb.

Values are also included in Table 1 for calorized specimens with the calorized layers removed, and from these it is evident that the strength and ductility characteristics are of the same order of

TABLE 3 CREEP CHARACTERISTICS OF CALORIZED 1.5 MOLYBDENUM STEEL AT VARIOUS TEMPERATURES

Temperature, F	Stress for creep rates of 0.01 and 0.1 per cent per 1000 hr, lb per sq in. At 0.01 per cent per 1000 hr	At 0.1 per cent per 1000 hr
1100	3350	10000
1150	3150	9400
1200	2100	7300
1300	1000	3350
1400	330	940

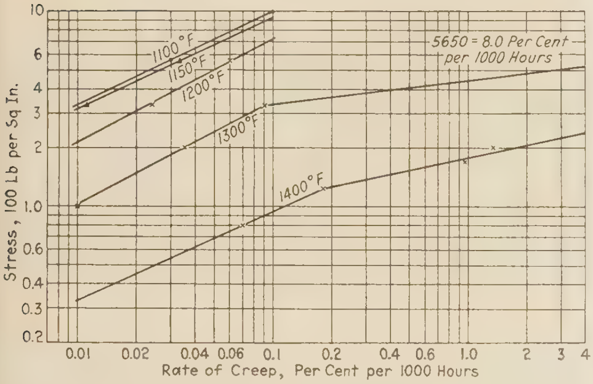

FIG. 2 CREEP-STRESS CURVES FOR CALORIZED 1.5 MOLYBDENUM STEEL

magnitude as the uncalorized material. This proves the calorizing to be without a detrimental influence on the properties of the base steel.

Air-Hardening Characteristics. In order to determine the susceptibility of this steel to air hardening, uncalorized cylinders were both air-cooled and water-quenched from a series of temperatures ranging from 1450 to 1750 F. The hardness values given in Table 2 were obtained on these cylinders.

In the case of the air-cooled specimens, the Brinell hardness ranged only from 137 to 153, indicating that this carbon-molybdenum steel is not susceptible to air-hardening.

Creep Characteristics. The creep characteristics of the calorized and heat-treated (annealed at 1650 F and drawn at 1300 F) steel were determined at 1100, 1150, 1200, 1300, and 1400 F. The single-step method of loading was used with four different stresses being employed at the two higher temperatures, and two stresses at the three remaining temperatures. Each test was continued for 1000 hr. All testing details were in accordance with the Tentative A.S.T.M. Standard E22-34T.

Fig. 1, setting forth the results obtained at 1300 F, is included as being representative of the actual time-elongation curves obtained at each of the five temperatures. If the stresses and corresponding second-stage creep rates are plotted to logarithmic coordinates, as shown in Fig. 2, stresses corresponding to definite creep rates can be obtained. Creep strength, as determined from this figure, is given in Table 3.

From these values it is evident that this analysis does possess very good load-carrying ability at temperatures up to and including 1300 F. Its superiority in this respect becomes even more evident if comparisons are made with other calorized analyses. For example, Figs. 3, 4, and 5 give time-elongation curves obtained for this analysis as well as for calorized 1.0-molybdenum steel under similar stresses at 1000, 1100, and 1400 F. It is apparent that the increase in the molybdenum content from 1.0 to 1.5 per cent has greatly improved the creep strength.

Tube design stress is often based upon some high percentage of the stress required for a creep rate of 1 per cent per 10,000 hr. Values to 1400 F are given since it is rather well-established that in both radiant-superheater and oil-still tube design, the ability of the tube to survive for several thousand hours at from 100 to 200 F above the normal temperature is more important than possession of requisite strength under normal operating conditions.

Proof of the last statement is had in the fact that failed oil-still tubes are occasionally found to exhibit internal carburization. Failed radiant-superheater tubes have also been observed with

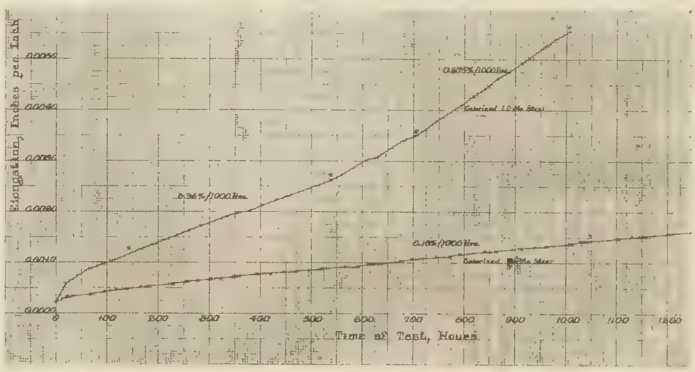

FIG. 4 TIME-ELONGATION CURVES FOR HEAT-TREATED CALORIZED STEELS TESTED AT 1100 F AND 10,000 LB PER SQ IN.

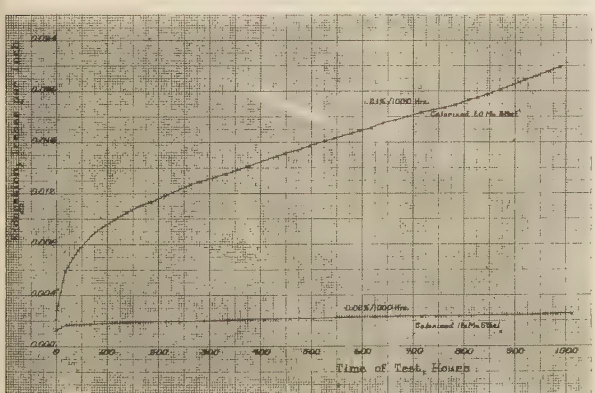

FIG. 3 TIME-ELONGATION CURVES FOR HEAT-TREATED CALORIZED STEELS TESTED AT 1400 F AND 2000 LB PER SQ IN.

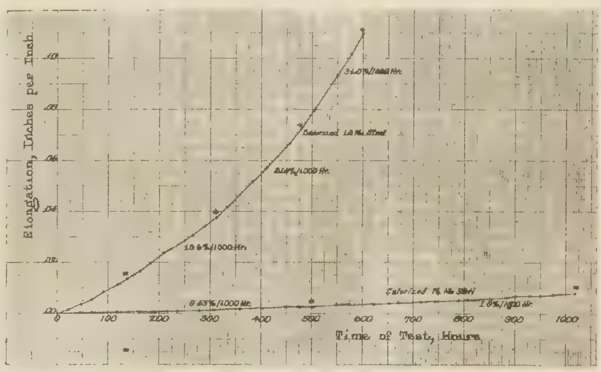

FIG. 5 TIME-ELONGATION CURVES FOR HEAT-TREATED CALORIZED STEELS TESTED AT 1000 F AND 24,000 LB PER SQ IN.

more scale on the inside of the tube than outside. Chemical analysis demonstrated this scale to be iron oxide (not boiler scale). It was due to dissociation of the steam into hydrogen and oxygen by contact with the tube wall at 1200 F. In the case of the oil still, these abnormal temperatures occur for some time at the end of every run because of the coke formation on the tube interior. In the case of radiant superheaters, the abnormal temperature comes

Fig. 6 Uncalorized and Calorized Specimens Tested at 1400 F and 2000 Lb per Sq In.

Specimen	Composition	Test time, hr	Fractured (F) or not fractured (NF)	Elongation, %
Uncalorized Specimen:				
A	0.10 C[a]	59.0	F	79.40
B	0.15 C + 0.50 Mo	110.0	F	45.70
C	0.15 C + 1.00 Mo	177.0	F	56.00
D	1.25 Cr + 0.50 Mo	230.0	F	21.70
E	1.50 Cr + 0.60 Mo	297.0	F	40.70
F	4–6 Cr + 0.50 Mo	487.0	F	40.10
Calorized Specimen:				
G	0.10 C[a]	99.5	F	41.20
H	0.15 C + 0.50 Mo	480.0	F	15.00
I	1.25 Cr + 0.50 Mo	600.0	NF	6.40
J	1.50 Cr + 0.50 Mo	600.0	NF	4.40
K	4–6 Cr + 0.50 Mo	600.0	NF	2.41
L	0.15 C + 1.00 Mo	600.0	NF	2.24
M	0.15 C + 1.50 Mo	600.0	NF	0.37

[a] Killed mild steel.

from reduced rate of flow under low loads or during shutting down and starting up periods.

The ability of the calorized 1.50 molybdenum steel to carry the load for hundreds of hours far above normal temperatures is shown in Fig. 6 illustrating tensile specimens tested at 2000 lb per sq in. at 1400 F. It is also demonstrated in Fig. 2. Choosing a stress, 3300 lb per sq in. for example, and following it across the various temperature lines in Fig. 2, it will be noted that the creep rate per 1000 hr at this stress is 0.01, 0.012, 0.024, and 0.09 per cent per 1000 hr at temperatures of 1100, 1150, 1200, and 1300 F, respectively. This implies that this steel would carry a load of 3300 lb per sq in. for 150,000 hr at a temperature of 1100 F with a total creep of $1^1/_2$ per cent; at the end of 10,000 additional hr at 100 F above a normal temperature of 1100 F the creep would be only 0.24 per cent; and at the end of 2000 hr additional at 200 F above a normal temperature of 1100 F the creep would be 0.18 per cent.

Let us assume hours in service at various temperatures as given in Table 4.

TABLE 4

Time, hr	Metal, temp F	Excess temp above normal, F	Creep per 1000 hr, per cent	Creep in time assumed (col. 1), per cent
150,000	1100	Normal	0.010	1.50
10,000	1200	100	0.024	0.24
2,000	1300	200	0.090	0.18
Total 162,000 hr ($18^1/_2$ yr)				Total 1.92 per cent

The relative load-carrying ability of a series of steels can also be determined by subjecting them to a given load at any given temperature and maintaining this load until a large number of them fracture. An examination of the resulting specimens will afford additional proof as to differences in their load-carrying ability. Fig. 6 shows specimens of several different steels which were heated for a maximum period of 600 hr at 1400 F under a stress of 2000 lb per sq in. Five of the calorized specimens did not fracture during this time but they did undergo varying amounts of extension, with the 1.5 molybdenum showing least deformation.

Applications

In general, the calorized 1.5 molybdenum steel has been used, thus far, for tubes, rods, and bolts. There are two partial installations of radiant superheaters and for one of these a most satisfactory operating report has been received.

There is one installation of liquid-phase cracking of very heavy cracked residuum oil heated to 1030 F with the unusually high rate of heat input of 18,000 Btu per sq ft per hr, which makes for high tube-wall temperatures. Another installation is used to heat petroleum vapor under pressure to 1400 F.

A quantity of calorized 1.5 molybdenum rods of $^1/_2$ in. diameter are in use to suspend insulating refractory brick (running through a hole in the brick) in an oil-still roof.

A heat-treating furnace for annealing brass and copper tubes uses 3 in. O.D. × 9 gage × 4 ft long calorized 1.5 molybdenum tubes suspended between two rows of alloy chain. The tubes to be annealed are carried through the furnace on the calorized tubes which serve as a slat conveyor.

None of the calorized tubes are adaptable to present designs of convection superheaters due to the impossibility of making either forged return ends or the very sharp return bends required. Radii of four to five times the tube diameter, used in radiant-superheater design, may be made in the calorized tube without injury to the surface alloy if bent hot.

The calorized molybdenum tube may be rolled into headers without damage to the calorized surface inside. The ends of the tubes are usually prepared by light grinding which removes about one half the calorized surface, thus reaching the more ductile alloy at the junction. By the use of specially coated rod of 25 per cent chromium and 12 per cent nickel, seal welding may be done on calorizing. If it is not necessary to make the weld oxidation-resisting, a less expensive job can be done by grinding off a band of calorizing around the tube where the bead is to come, and welding with coated steel rod.

Conclusions

On the basis of the results obtained it is believed that calorized 1.5 per cent molybdenum steel is well-suited for service at elevated temperatures. Its room-temperature physical properties and its lack of susceptibility to air hardening insure freedom from difficulty during installation. The calorized surface imparts a very marked resistance to oxidation and sulphur corrosion and the high creep strength of the base steel prevents the calorized surface from cracking. In general, no cracking of this surface will occur until the deformation exceeds 5.0 per cent.

Power and Forces in Milling S.A.E. 3150 Steel With Helical Mills

By O. W. BOSTON,[1] W. W. GILBERT,[2] and K. B. KAISER,[3] ANN ARBOR, MICH.

This paper outlines a series of tests conducted on a milling machine using slab mills of similar construction except for variation in rake angle, when milling an S.A.E. 3150 steel. The cutters had rake angles of 20, 15, 10, and 5 deg as variables, but otherwise were 3 in. in diameter, and had 12 teeth on a 25-deg left-hand helix. Gross and net power, as determined on a wattmeter, and the three components of the cutting force, as determined on a specially constructed dynamometer, were obtained for a wide variety of combinations of depths of cut and feed. The speed was constant, and a single cutting fluid was used in all tests.

From the data presented graphically, formulas for net power input, the horsepower per cubic inch per minute, and for the tangential and axial components of the cutting force have been determined for each cutter as a function of feed and depth of cut. For any size of cut the tangential and normal components are shown to decrease as the rake angle is increased. The axial component shows comparatively little change in value up to a 10-deg rake angle, after which it decreases rapidly for an increase up to 20 deg. The horsepower per cubic inch per minute decreases rapidly with an increase in rake angle.

For each cutter the tangential and axial components increase as the feed or depth of cut is increased. The normal component compresses the work for small values of feed and depth of cut but tends to lift the work for large values. Gross and net values of horsepower for the 15-deg rake angle cutter are plotted for each of a number of values of depth of cut over feed as a variable. From these curves, values may be selected for any combination of depth of cut and feed.

Similar data are included, also, for a cutter having eight teeth, 45-deg helix angle, and 15-deg rake angle.

THIS PAPER presents experimental data showing the effect of variable feed, depth of cut, rake angle, and helix angle upon the power as determined by a wattmeter and the three components of the cutting force as determined by a dynamometer when milling an annealed S.A.E. 3150 steel. A new Cincinnati No. 4 plain milling machine was used. It was equipped with a direct-connected 15-hp motor, to which was attached a recording wattmeter, as shown in Fig. 1. The wattmeter record gave the tare power required to operate the machine running idly with the spindle and feed engaged and the gross power when taking the actual cut in each case, so that the net power developed at the cutter could be computed. In all tests a cutting fluid was circulated at the rate of approximately five gallons a minute to the cutter by an independent circulating system shown at the lower left of the machine in Fig. 1. The cutting fluid consisted of 1 part of sulphurized-base oil blended with 15 parts of light mineral oil having a viscosity of 110 sec Saybolt at 100 F.

The dynamometer was of an all-steel type employing the principle that the deflection of a steel beam is proportional to the component of the cutting force. Its construction might be compared with that of a weighing scale which measures both positive and negative forces. It consisted of three independent beam structures, each connected through lever mechanisms to separate dial indicators. In this way, the three components of the cutting force, that is, N normal to the machined surface, T tangential to the machined surface, and A axial (side thrust) were measured. The dynamometer was calibrated with a small hydraulic unit provided with a pressure gage, which had a range up to 3000 lb. The unit was braced against the dynamometer in the direction of the component being calibrated, and the force applied by a screw jack or by using the table-feed mechanism. The force exerted caused a deflection of the dynamometer as indicated by the dial gage. A deflection of 0.001 in. on the respective dial gages indicated normal, tangential, and axial forces of 21, 14, and 5.75 lb. Actual cutting forces were obtained by multiplying each dial reading by its constant. Calibration was checked be-

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[2] Instructor, College of Engineering, University of Michigan. Jun. A.S.M.E. Dr. Gilbert was graduated in mechanical engineering from the University of Colorado in 1931, received his master's degree in 1932, and his Sc.D. degree in 1935 from the University of Michigan. Since January, 1932, Dr. Gilbert has worked with Professor Boston on various phases of metal cutting and is coauthor of several metal-cutting papers.

[3] Ingersoll Milling Machine Company, Rockford, Ill. Mr. Kaiser was graduated in mechanical engineering from the University of Michigan in February, 1937. Prior to his graduation he was an assistant for two years to Professor Boston.

[4] "Performance of Cutting Fluids," by O. W. Boston and C. J. Oxford, Trans. A.S.M.E., vol. 54, 1932, paper MSP-54-2. Also: "Performance of Cutting Fluids in Drilling Various Metals," by O. W. Boston and C. J. Oxford, Trans. A.S.M.E., vol. 55, 1933, paper RP-55-1. Also: "Performance of Cutting Fluids When Sawing Various Metals," by O. W. Boston and C. E. Kraus, Trans. A.S.M.E., vol. 56, 1934, paper RP-56-5. Also: "Elements of Milling, Part 2," by O. W. Boston and C. E. Kraus, Trans. A.S.M.E., vol. 56, 1934, paper RP-56-1.

[5] "Elements of Milling," by O. W. Boston and C. E. Kraus, Trans. A.S.M.E., vol. 54, 1932, paper RP-54-4; also, Trans. A.S.M.E., vol. 56, 1934, paper RP-56-1, pp. 355–371.

Contributed by the Special Research Committee on Cutting of Metals and presented at the Fall Meeting of The American Society of Mechanical Engineers, held at Erie, Pa., Oct. 4–6, 1937.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1937, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

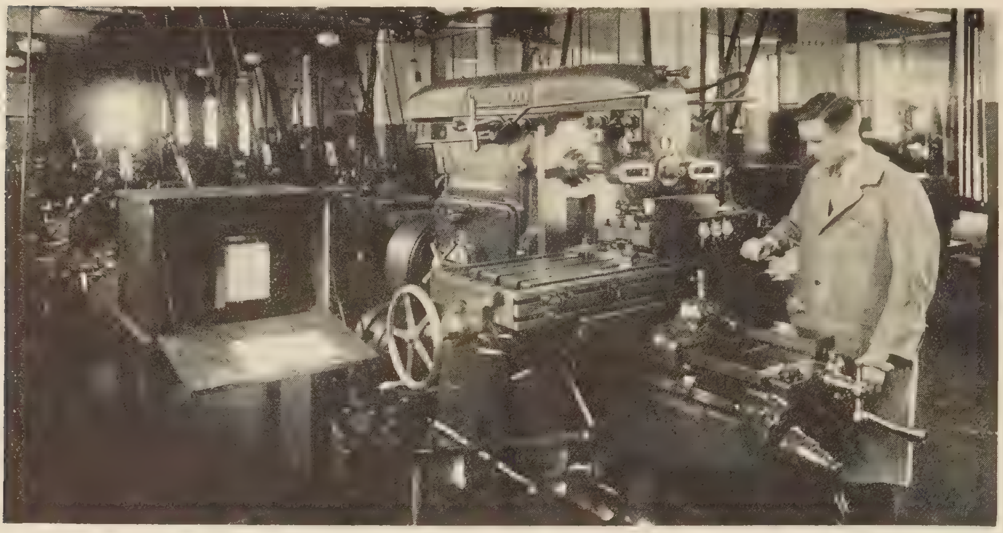

Fig. 1 The Milling Machine, Wattmeter, and Dynamometer Used in the Tests

fore and after the tests on each cutter to insure accuracy of results.

The cutters used were of high-speed steel and specially made to provide the various rake and helix angles and number of teeth desired. These cutters are listed in Table 1. Cutters Nos. 1 to 4 were 3 in. in diameter by 3 in. long, and had 12 teeth on 25-deg left-hand helixes and 7-deg relief angles. These cutters varied only in rake angle, having 20, 15, 10, and 5 deg, respectively. Cutter No. 5 had 8 teeth on a 45-deg left-hand helix angle and a 15-deg rake angle. The material cut consisted of hot-rolled annealed S.A.E. 3150 steel in the form of slabs 10 in. long, 6 in. high, and 2 in. wide. This material, used in other tests[4] (see footnote 4 on p. 545), had a Brinell hardness of 170, and was found to be very uniform throughout.

The Tests

Before taking force and power readings, a variety of cuts were taken at various combinations of speed, feed, and depth to obtain an idea as to the performance of the dynamometer. The speed of 13.3 fpm proved to be most satisfactory for the combinations of feed and depth of cut as reported in Table 1, in that no chatter or excessive fluctuation of the gage indicators occurred. At higher speeds some trouble was experienced. In running the tests for each cutter, the speed and depth of cut were held constant, while the feed was varied over each traverse of the block; then the next depth of cut was selected and another traverse of the block made, using the same variable feeds. Four separate tests were run for each combination of feed and depth of cut. In a few cases where values did not check satisfactorily, further tests were made. All tests were run in the conventional or up-milling manner. The width of all cuts was 2 in., and the forces shown on the charts are for this width.

All data for the 20-deg rake, 25-deg helix angle cutter are presented in Figs. 2 to 7, inclusive; for the 15-deg rake angle cutter in Figs. 8 to 13, inclusive; for the 10-deg rake angle cutter in Figs. 14 to 19, inclusive; and for the 5-deg rake angle cutter in Figs. 20 to 25, inclusive. Data for the 8-tooth, 15-deg rake, 45-deg helix angle cutter are given in Figs. 26 to 31, inclusive.

Values of the three components and net horsepower per cubic inch per minute are summarized for each cutter for one size of cut in Fig. 32, and the gross and net power values for all cuts for the 15-deg rake, 25-deg helix cutter are summarized in Fig. 33. The cutters are listed, and equations representing the components and net horsepower per cubic inch per minute as a function of the depth of cut and feed are summarized in Table 1.

It may be pointed out that the feed per tooth f for a feed F of 1

TABLE 1 SUMMARY OF FORCE AND POWER TESTS WHEN MILLING AN ANNEALED S.A.E. 3150 STEEL

Cutter No.	Cutter Description (Left helix slab mill)	Cutting speed Rpm	Cutting speed Fpm	Range of cuts: Depth, in.	Range of cuts: Feed, in.	Axial force (side thrust)	Tangential force	Net horsepower per cu in. per min[b]
1	3″ diam × 3″ long, 12 teeth, 25-deg helix, 20-deg rake	17	13.3			$8{,}820\ wd^{0.683} f^{0.65}$ 140[a]	$72{,}600\ wd^{0.77} f^{0.688}$ 770	$\frac{0.469}{d^{0.175} f^{0.175}}$ 1.70
2	3″ diam × 3″ long, 12 teeth, 25-deg helix, 15-deg rake	17	13.3			$9{,}960\ wd^{0.655} f^{0.67}$ 155	$81{,}150\ wd^{0.80} f^{0.690}$ 800	$\frac{0.500}{d^{0.130} f^{0.190}}$ 1.715
3	3″ diam × 3″ long, 12 teeth, 25-deg helix, 10-deg rake	17	13.3	0.035 to 0.200	$^3/_4$–$3^3/_8$ per min; 0.00368 to 0.01654 per tooth	$10{,}960\ wd^{0.623} f^{0.695}$ 164	$95{,}350\ wd^{0.83} f^{0.695}$ 840	$\frac{0.552}{d^{0.086} f^{0.200}}$ 1.77
4	3″ diam × 3″ long, 12 teeth, 25-deg helix, 5-deg rake	17	13.3			$11{,}900\ wd^{0.605} f^{0.725}$ 162	$107{,}000\ wd^{0.87} f^{0.686}$ 910	$\frac{0.591}{d^{0.042} f^{0.216}}$ 1.81
5	3″ diam × $2^1/_2$″ long, 8 teeth, 45-deg helix, 15-deg rake	26	20.4		$^3/_4$–$3^3/_8$ per min; 0.00361 to 0.0162 per tooth	$23{,}500\ wd^{0.840} f^{0.70}$ 195	$77{,}700\ wd^{0.920} f^{0.715}$ 495	$\frac{0.74}{d^{0.084} f^{0.120}}$ 1.62

[a] Value for d = 0.075 in., f = 0.009 in. per tooth, and w = 2 in. (See Fig. 3.)
[b] Net horsepower equals the gross wattmeter power less the tare for machine running idly.

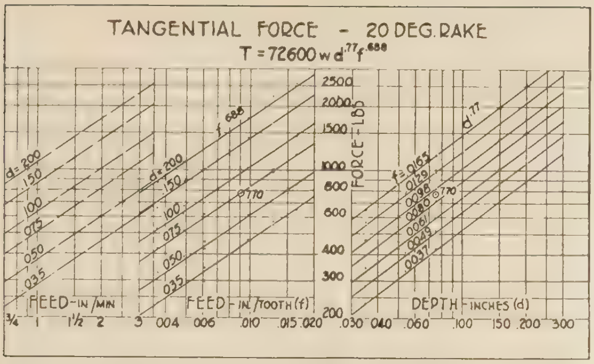

Fig. 2 Tangential Forces Plotted on Log-Log Paper for the 20-Deg Rake, 25-Deg Left-Hand Helix Slab Mill 3 In. in Diameter, Having 12 Teeth, Operating at 17 Rpm (13.3 Fpm), When Cutting Annealed S.A.E. 3150 Steel at Various Feeds and Depths With a Sulphurized Mineral-Lard Oil, Width of Cut 2 In.

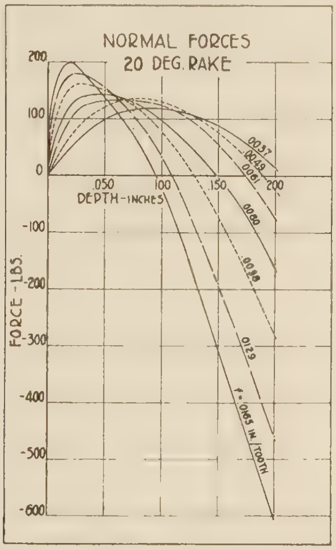

Fig. 4 Normal Forces Plotted on Cartesian Coordinates Over the Depth of Cut in Inches for the Cuts Described in Fig. 2

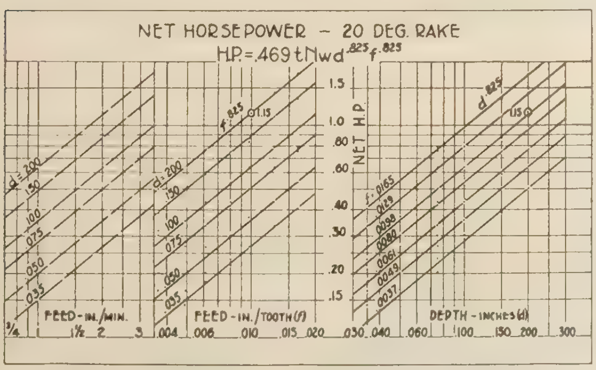

Fig. 6 Net Horsepower as Determined From the Wattmeter for Various Feeds and Depths Plotted on Log-Log Paper for the Cuts Described in Fig. 2

t = number of teeth in the cutter
N = speed, rpm
w = width of cut, in.
d = depth of cut, in.
f = feed per tooth, in.

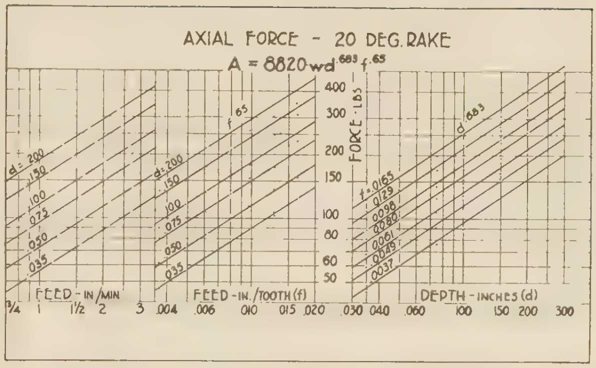

Fig. 3 Axial Forces Plotted on Log-Log Paper for the Cuts Described in Fig. 2

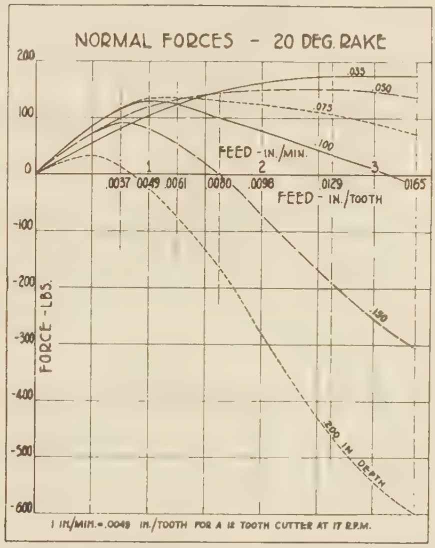

Fig. 5 Normal Forces Plotted on Cartesian Coordinates Over the Feed in Inches per Tooth and Inches per Minute for the Cuts Described in Fig. 2

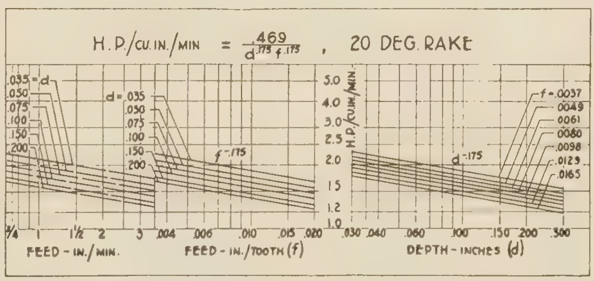

Fig. 7 Net Horsepower per Cubic Inch per Minute Plotted Over Feed and Depth for the Cuts Described in Fig. 2

in. per min varies as t the number of teeth in the cutter and n the revolutions per minute.

Then

$$F = ftn$$

When

F is 1 in. for the 12-tooth cutter operating at 17 rpm

$$1 = f \times 12 \times 17$$

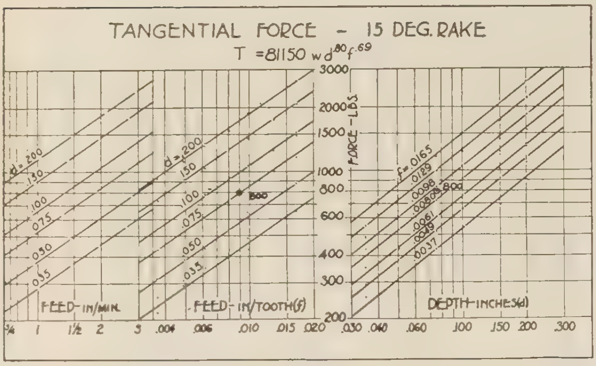

Fig. 8 Tangential Forces Plotted on Log-Log Paper for the 15-Deg Rake, 25-Deg Left-Hand Helix Slab Mill 3 In. in Diameter, Having 12 Teeth, Operating at 17 Rpm (13.3 Fpm), When Cutting Annealed S.A.E. 3150 Steel at Various Feeds and Depths With a Sulphurized Mineral-Lard Oil, Width of Cut 2 In.

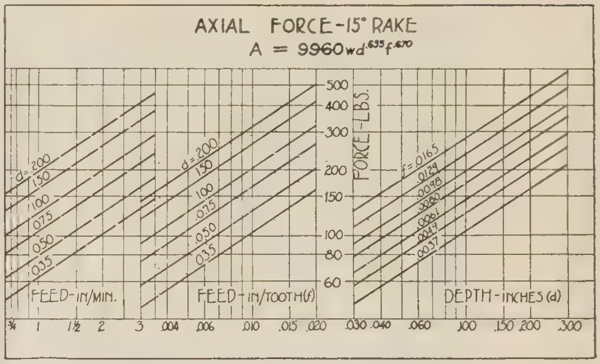

Fig. 9 Axial Forces Plotted on Log-Log Paper for the Cuts Described in Fig. 8

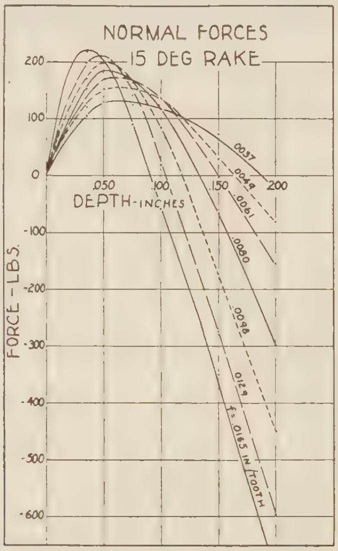

Fig. 10 Normal Forces Plotted on Cartesian Coordinates Over the Depth of Cut in Inches for the Cuts Described in Fig. 8

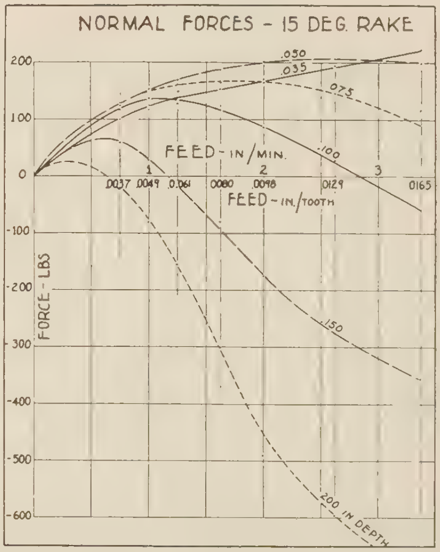

Fig. 11 Normal Forces Plotted on Cartesian Coordinates Over the Feed in Inches per Tooth and Inches per Minute for the Cuts Described in Fig. 8

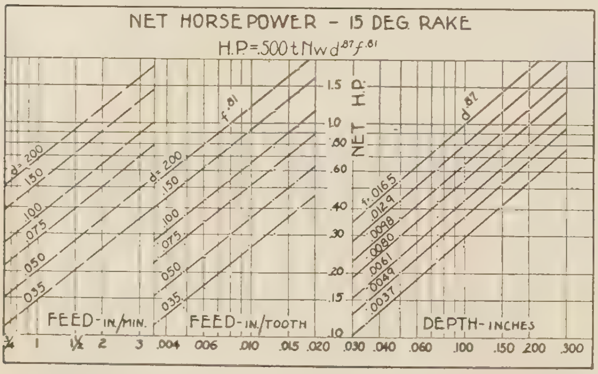

Fig. 12 Net Horsepower as Determined From the Wattmeter for Various Feeds and Depths Plotted on Log-Log Paper for the Cuts Described in Fig. 8

t = number of teeth in the cutter
N = speed, rpm
w = width of cut, in.
d = depth of cut, in.
f = feed per tooth, in.

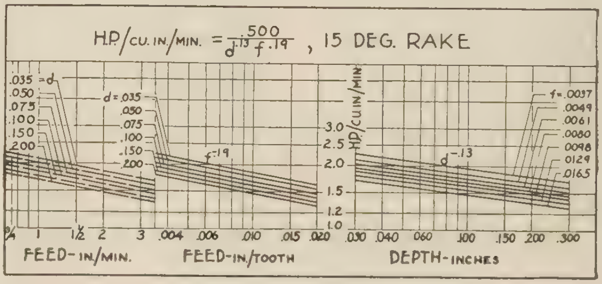

Fig. 13 Net Horsepower per Cubic Inch per Minute Plotted Over Feed and Depth for the Cuts Described in Fig. 8

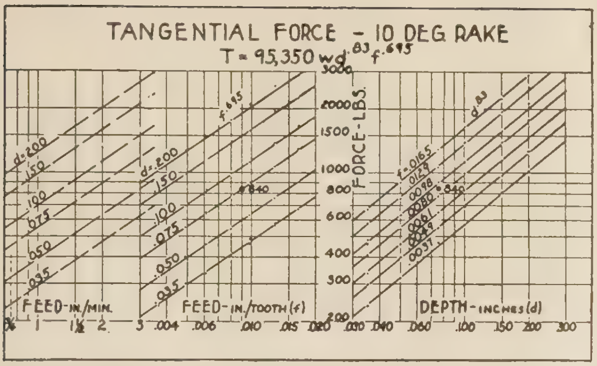

Fig. 14 Tangential Forces Plotted on Log-Log Paper for the 10-Deg Rake, 25-Deg Left-Hand Helix Slab Mill 3 In. in Diameter, Having 12 Teeth, Operating at 17 Rpm (13.3 Fpm), When Cutting Annealed S.A.E. 3150 Steel at Various Feeds and Depths With a Sulphurized Mineral-Lard Oil, Width of Cut 2 In.

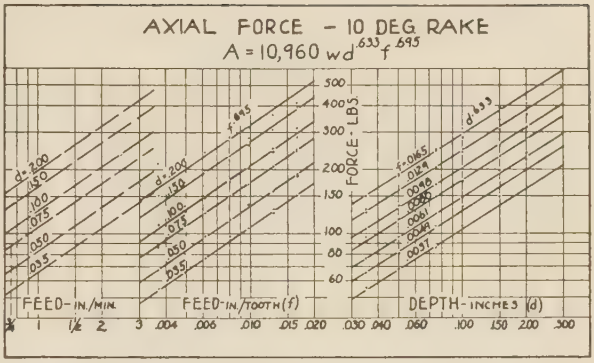

Fig. 15 Axial Forces Plotted on Log-Log Paper for the Cuts Described in Fig. 14

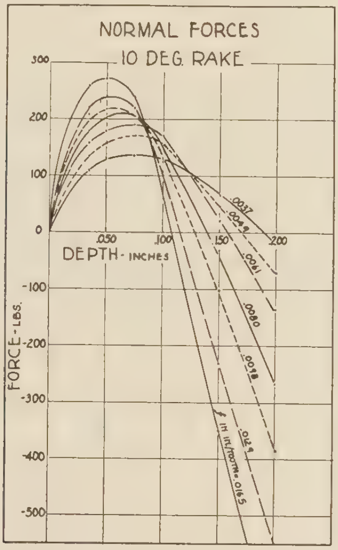

Fig. 16 Normal Forces Plotted on Cartesian Coordinates Over the Depth of Cut in Inches for the Cuts Described in Fig. 14

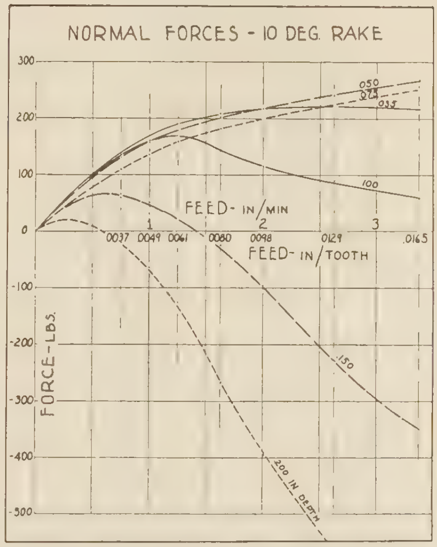

Fig. 17 Normal Forces Plotted on Cartesian Coordinates Over the Feed in Inches per Tooth and Inches per Minute for the Cuts Described in Fig. 14

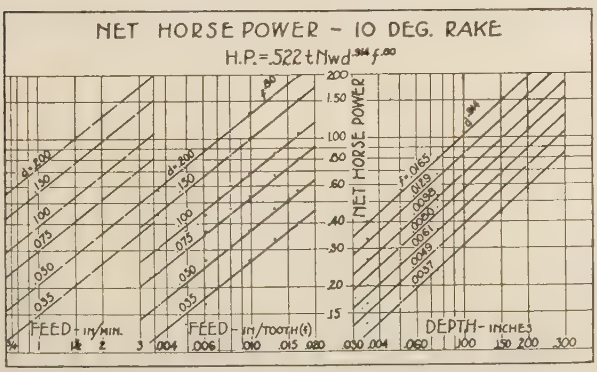

Fig. 18 Net Horsepower as Determined From the Wattmeter for Various Feeds and Depths Plotted on Log-Log Paper for the Cuts Described in Fig. 14

t = number of teeth in the cutter
N = speed, rpm
w = width of cut, in.
d = depth of cut, in.
f = feed per tooth, in.

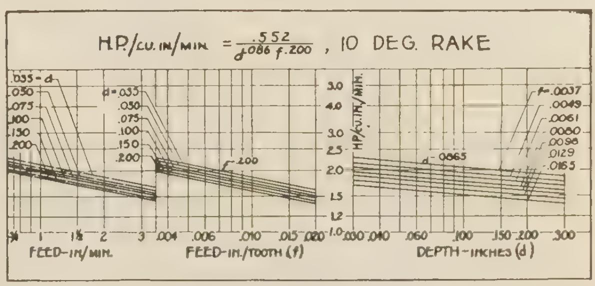

Fig. 19 Net Horsepower per Cubic Inch per Minute Plotted Over Feed and Depth for the Cuts Described in Fig. 14

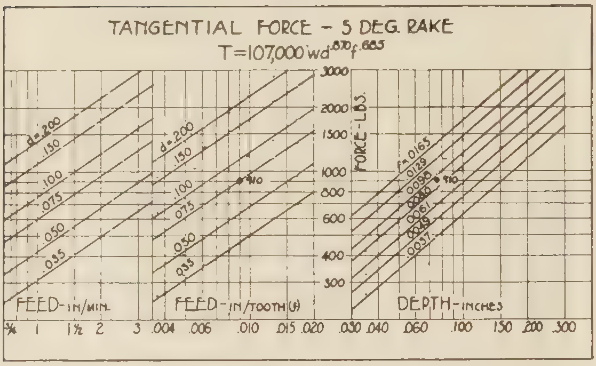

Fig. 20 Tangential Forces Plotted on Log-Log Paper for the 5-Deg Rake, 25-Deg Left-Hand Helix Slab Mill 3 In. in Diameter, Having 12 Teeth, Operating at 17 Rpm (13.3 Fpm), When Cutting Annealed S.A.E. 3150 Steel at Various Feeds and Depths With Sulphurized Mineral-Lard Oil, Width of Cut 2 In.

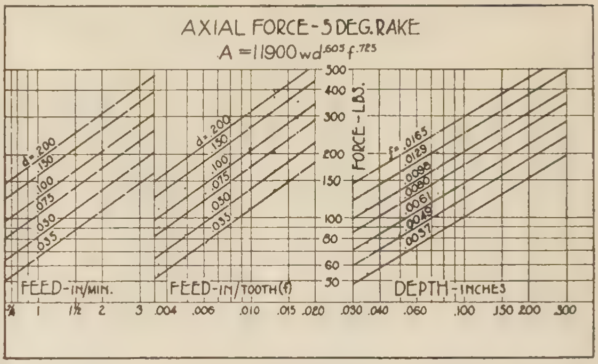

Fig. 21 Axial Forces Plotted on Log-Log Paper for the Cuts Described in Fig. 20

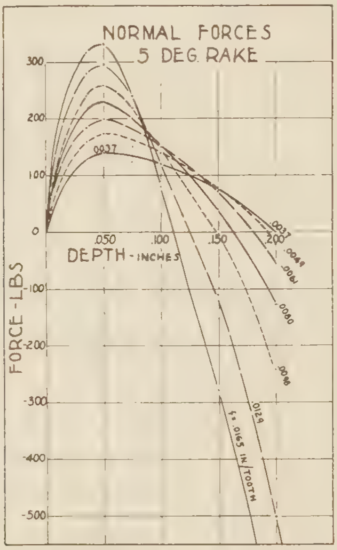

Fig. 22 Normal Forces Plotted on Cartesian Coordinates Over the Depth of Cut in Inches for the Cuts Described in Fig. 20

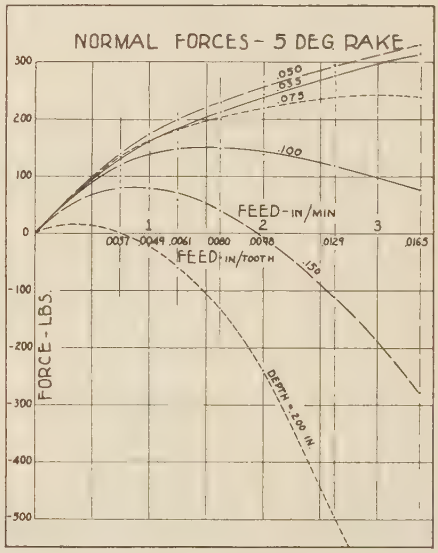

Fig. 23 Normal Forces Plotted on Cartesian Coordinates Over the Feed in Inches per Tooth and Inches per Minute for the Cuts Described in Fig. 20

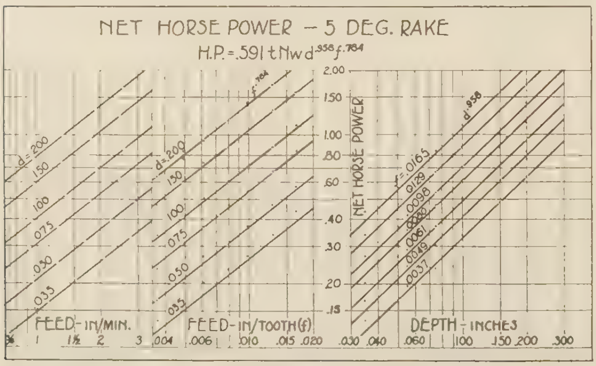

Fig. 24 Net Horsepower as Determined From the Wattmeter for Various Feeds and Depths Plotted on Log-Log Paper for the Cuts Described in Fig. 20

t = number of teeth in the cutter
N = speed, rpm
w = width of cut, in.
d = depth of cut, in.
f = feed per tooth, in.

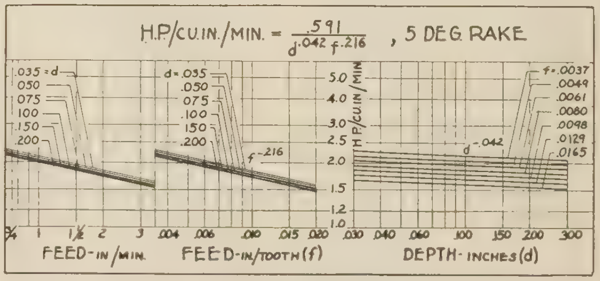

Fig. 25 Net Horsepower per Cubic Inch per Minute Plotted Over Feed and Depth for the Cuts Described in Fig. 20

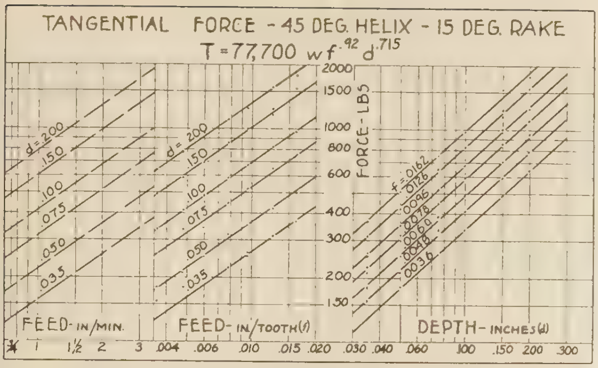

Fig. 26 Tangential Forces Plotted on Log-Log Paper for a 15-Deg Rake, 45-Deg Left-Hand Helix Slab Mill 3 In. in Diameter, Having 8 Teeth, Operating at 26 Rpm (20.4 Fpm), When Cutting Annealed S.A.E. 3150 Steel at Various Feeds and Depths With a Sulphurized Mineral-Lard Oil, Width of Cut 2 In.

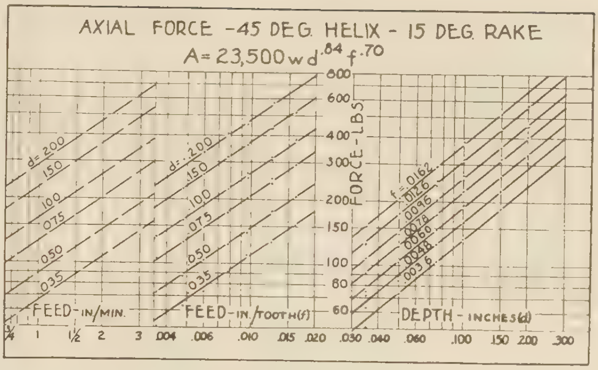

Fig. 27 Axial Forces Plotted on Log-Log Paper for the Cuts Described in Fig. 26

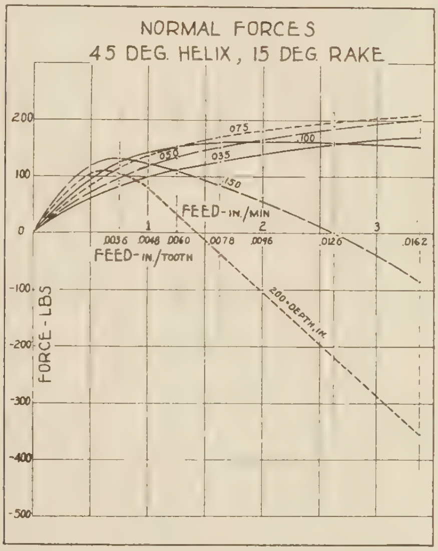

Fig. 28 Normal Forces Plotted on Cartesian Coordinates Over the Feed in Inches per Tooth and Inches per Minute for the Cuts Described in Fig. 26

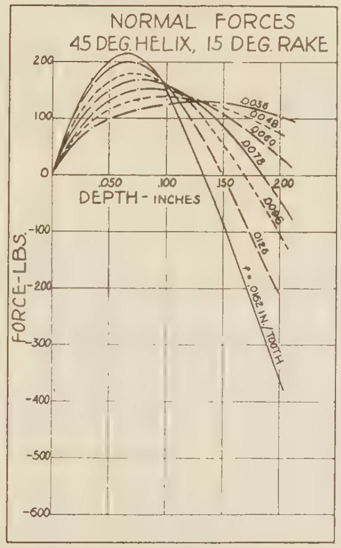

Fig. 29 Normal Forces Plotted on Cartesian Coordinates Over Depth of Cut in Inches for the Cuts Described in Fig. 26

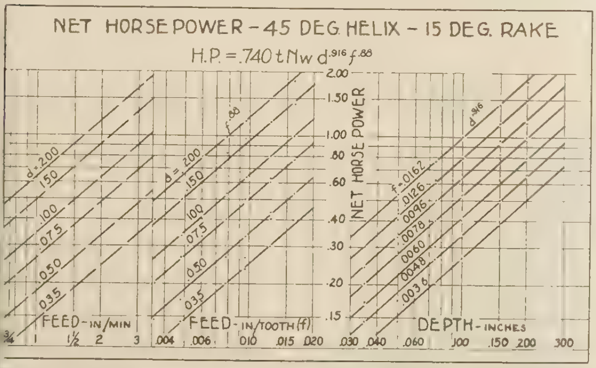

Fig. 30 Net Horsepower as Determined From the Wattmeter for Various Feeds and Depths of Cut Plotted on Log-Log Paper for the Cuts Described in Fig. 26

t = number of teeth in the cutter
N = speed, rpm
w = width of cut, in.
d = depth of cut, in.
f = feed per tooth, in.

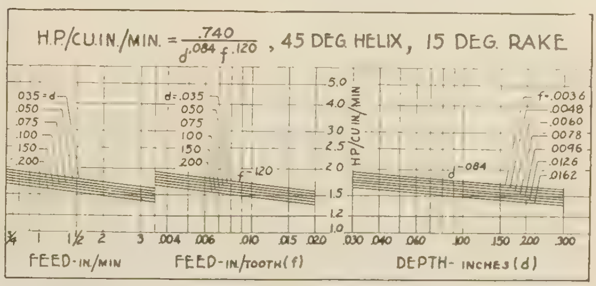

Fig. 31 Net Horsepower per Cubic Inch per Minute Plotted Over Feed and Depth of Cut for the Cuts Described in Fig. 26

or $f = 0.0049$ in. as shown in Fig. 5. For the 8-tooth cutter operating at 26 rpm

$$1 = f \times 8 \times 26$$

or $f = 0.0048$ in. as shown in Fig. 28. The feed per tooth is practically the same for the two cutters, and yet the tangential force for the 12-tooth, 15-deg rake angle, 25-deg helix angle cutter for a cut 2 in. wide, as shown in Table 1, is 800 lb, but only

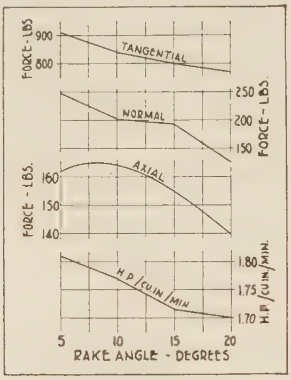

Fig. 32 Summary of Forces and Net Horsepower per Cubic Inch per Minute for the 3-In. Diameter, 12-Tooth, 25-Deg Helix Slab Mill Having Rake Angles as Indicated, When d = 0.750 In. and f = 0.0090 In. per Tooth Cutting S.A.E. 3150 Steel at 17 Rpm (13.3 Fpm), With the Sulphurized Mineral-Lard Oil

495 lb for the 8-tooth, 15-deg rake angle, 45-deg helix angle cutter.

Tests With the 12-Tooth, 20-Deg Rake Angle Cutter. The tangential forces are shown on logarithmic coordinates in Fig. 2 for cutter No. 1, having the 20-deg rake angle for a width of cut of 2 in. At the right, the forces are plotted over the depth of cut in inches as the variable for each of several feeds. These data for each value of feed give straight lines which are parallel and have a slope with the horizontal of an angle whose tangent is 0.77. This slope becomes the exponent of the variable d. The forces also are plotted over the variable feed in inches per tooth, in the center of Fig. 2, for each of several depths of cut. These lines are parallel and have a slope from the horizontal, the tangent of which is 0.688. This becomes the exponent of the variable f. These straight lines on logarithmic coordinates, representing cutting force over variable depth and feed, conform to the general equation

$$F = Kwd^x f^y$$

in which

F = force, lb
K = a constant depending upon the material cut and the cutter for this speed
W = width of cut, in.
d = depth of cut, in.
f = feed, in. per tooth

The equation for the tangential force T then is

$$T = 72{,}600wd^{0.77}f^{0.688}$$

for this cutter, as shown with the other formulas in Table 1.

Illustrative Problem. Find the tangential force on the milling cutter 3 in. in diameter having 12 teeth, 20-deg rake angle and 25-deg helix angle, operating at 17 rpm when taking a cut in the annealed S.A.E. 3150 steel 2 in. wide, 0.075 in. deep, and 0.0098 in. per tooth feed. The logarithmic form of the above equation is

$$\log T = \log 72{,}600 + \log 2 + 0.77 (\log 0.075) + 0.688 (\log 0.0098)$$

From the logarithmic table, the mantissa of 72,600 is 0.86101, but as the number has five digits, the log has a characteristic of 4, resulting in 4.86101. Also

$$0.77 (\log 0.075) = 0.77 (-2.87506) = 0.77 (8.87506 - 10) = 6.83380 - 7.7$$

and

$$0.688 (\log 0.0098) = 0.688 (-3.99123) = 0.688 (7.99123 - 10) = 5.49795 - 6.88$$

Then

$$\log T = 4.86101 + 0.30103 + 6.83380 - 7.7 + 5.49795 - 6.88 = 2.91379$$

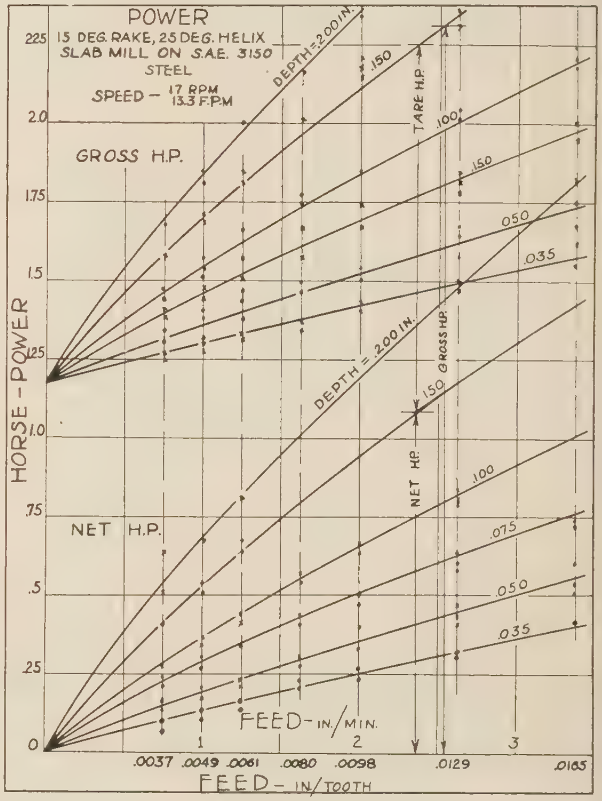

Fig. 33 Gross and Net Power for Various Feeds and Depths of Cut for the 15-Deg Rake, 12-Tooth Helical Mills

The number whose mantissa is 0.91379 is 82, but the characteristic 2 indicates three digits, so T = 820 lb.

From Fig. 2 the tangential force on this cutter may be read for any combination of feed and depth and is shown to be 770 lb for a width of 2 in., a depth of cut of 0.075 in., and a feed of 0.009 in. per tooth, as recorded in Table 1. This tangential component is of most importance in determining the power required in slab milling when shallow cuts are taken. For convenience, the forces are plotted again at the extreme left of Fig. 2 over the feed in inches per minute for the various values of depth of cut.

It is seen from this formula that, for any depth of cut, the tangential force increases with an increase in feed but at a lower rate. For example, for a depth of cut of 0.075 in., the force is 445 lb for a feed of 0.004 in. and a width of 2 in., but is only 710 lb for a feed of 0.008 in. The feed was increased 100 per cent.

while the force increased only 60 per cent. Similarly, for a feed of 0.008 in. per tooth, the force is 885 lb for a depth of cut of 0.100 in. and a width of 2 in., and is only 1500 lb, a 70 per cent increase, for a depth of 0.200 in. This shows why heavy feeds and depths of cut should be used from a power-economy point of view and that an increase in feed is more economical than an increase in depth for this cutter.

Corresponding values of axial (side-thrust) forces are plotted over depth of cut as the variable at the right in Fig. 3 and over the feed in inches per tooth in the center of the figure for the 2-in-wide cut. The slope of the variable-depth curve is 0.683, and of the variable-feed curve is 0.65, giving rise to the axial component equation

$$A = 8820\ wd^{0.683}f^{0.65}$$

as shown in Table 1.

Normal forces acting on the 20-deg rake angle cutter are shown plotted over depth of cut as a variable for each of seven values of feed on Cartesian coordinates in Fig. 4. The resulting curves have been found to conform to no simple mathematical equation and therefore are represented only graphically. For the lightest feed of 0.0037 in. per tooth (3/4 in. per min), the normal force is shown to increase rapidly with an increase in depth of cut up to 0.075 in., to remain practically constant to 0.125 in., and then to fall off to zero for a depth of cut slightly in excess of 0.200 in. The normal forces for heavier feeds increase rapidly at shallow depths to a maximum and then fall off and become negative for the greater values of depth. Values reported here are considered of importance because of their influence on the design and construction of machine tools and fixtures. The normal-force curves are replotted on Cartesian coordinates in Fig. 5 for various values of depth as the constant over the feed as the variable. The curves of this figure again are seen to be irregular and fall below the zero-axis for the larger values of depth of cut.

Values of net horsepower have been computed from the gross and tare values of power, as taken from the wattmeter charts for the 20-deg rake cutter shown as No. 1 in Table 1. These values are shown plotted on log-log paper for various values of feed over the depth of cut in inches at the right in Fig. 6. A series of straight parallel lines is obtained which make an angle with the horizontal whose tangent is 0.825. Net horsepower values are then plotted for various depths of cut over the feed in inches per tooth as the variable in the center of Fig. 6. Again a series of straight parallel lines is obtained, the slope from the horizontal of which corresponds to an angle whose tangent is 0.825. This then would give an equation for net horsepower for this 20-deg cutter of

$$Hp = 0.469\ tnwf^{0.825}d^{0.825}$$

For convenience, the values of net horsepower are plotted over the feed in inches per minute for different values of depth of cut at the left in Fig. 6. From Fig. 6 the net horsepower for slab-milling the S.A.E. 3150 steel is shown to be 1.15 for a depth of cut of 0.200 in., a feed of 0.010 in., and a width of 2 in.

Net horsepower to remove 1 cu in. per min has been computed and is plotted over the depth of cut in inches as the variable at the right in Fig. 7 and over the feed at the left. The slopes from the horizontal of the lines for various constant feeds over depth of cut as the variable correspond to an angle whose tangent is -0.175. Also, the values for various constant depths plotted over feed in inches per tooth have a negative slope corresponding to 0.175. From these two sets of curves, the equation for net horsepower per cubic inch per minute is

$$\text{Hp per cu in. per min} = 0.469/d^{0.175}f^{0.175}$$

It is seen that, as the feed is increased 346 per cent from 0.0037 to 0.0165 in. per tooth for a constant depth of cut of 0.200 in., the power per cubic inch of metal removed per minute is reduced from 1.67 to 1.27, or 24 per cent. Also, it is seen from the central part of Fig. 7, for a feed of 0.010 in. per tooth, that, if the depth of cut is increased from 0.035 to 0.200 in., or 470 per cent, the horsepower per cubic inch per minute is reduced from 1.89 to 1.4, or 26 per cent.

Tests With the 12-Tooth, 15-, 10-, and 5-Deg Rake Angle Cutters. Various force and power values obtained with the 15-deg rake, 25-deg helix angle cutter are shown in detail in Figs. 8 to 13, inclusive. Equations in each case are developed which are similar to those for the 20-deg rake cutter, as shown on the graphs and summarized in Table 1. Figs. 14 to 19, inclusive, show the data obtained with the 10-deg rake, 25-deg helix angle cutter, and Figs. 20 to 25, inclusive, show those obtained with the 5-deg rake, 25-deg helix angle cutter.

These data, obtained with cutters of different rake angles, show that cutters having the high values of rake angles are decidedly superior to those of low rake angle, when considering forces and power consumption. The 5-deg rake angle cutter produced higher force values and required more power to drive it than the 20-deg rake angle cutter. These cutters may be compared for a light cut of 0.075 in. depth and 0.009 in. feed per tooth as follows: the 5-deg rake angle cutter requires 15.7 per cent more axial force, 18.1 per cent more tangential force, 72 per cent more normal force, and 6.4 per cent more power per cubic inch of metal cut per minute than the 20-deg rake angle cutter. Actual values of these forces are given below the formulas in Table 1. It is interesting to note that this increase in horsepower per cubic inch per minute is less than the increase in tangential force which, for light depths of cut, usually controls the power input. In this case, however, the large increase in the normal force had the effect of changing the direction of the force component which resulted in the smaller power increase.

When comparing these cutters at the heaviest cuts of 0.0165 in. feed per tooth and 0.200 in. depth, it was found that the 20-deg rake cutter was even more superior than when based on the light cut. With heavy feeds and depths of cut the 5-deg rake cutter required 17.5 per cent more axial force, 24 per cent more tangential force, 24 per cent more power per cubic inch per minute, and the normal force was —750 lb for the 5-deg rake and only —600 lb for the 20-deg rake cutter. These force values, except for the normal forces, can be represented by mathematical equations, which are summarized in Table 1. It should be noted that not only are the constants of the equation changed, but also the exponents of the feed and depth are varied as the rake angle is changed from 20 to 5 deg.

For tangential and axial forces the lowest values of the exponents are most favorable, since they indicate more efficient removal of metal when taking heavy cuts. With the exception of the depth exponent for the axial force, all other exponents for these forces show favorable changes when the rake angle is changed from 5 to 20 deg.

Tests With the 45-Deg Helix Angle, Eight-Tooth, and 15-Deg Rake Angle Cutter. To determine the influence of helix, further tests were run with cutter No. 5. Data obtained with this cutter, which had 8 teeth, a 15-deg rake angle, and a 45-deg helix angle, are shown in Figs. 26 to 31, inclusive. All equations, except those for net horsepower, are summarized in Table 1.

In order that the same number of chips would be removed per minute with the 8-tooth cutter as with the 12-tooth previously reported on, cutting speed was increased by approximately 12/8 to give 26 rpm, or 20.4 fpm, as indicated in Table 1. These speeds were as close to the desired ratio as could be obtained by the change gears. While feeds in inches per minute were kept the same with the 8-tooth cutter as with the 12-tooth, the slight variation from the 12/8 ratio of speed gave a new series of feeds

in inches per tooth of 0.00361 as a minimum, compared with 0.00368 for the 12-tooth cutter. Under these conditions, it is felt that the difference in performance of cutter No. 5, having 8 teeth, 45-deg helix angle, and 15-deg rake angle, varies from that of cutter No. 2, having 12 teeth, 25-deg helix angle, and 15-deg rake angle, as a result of the change in helix. The small change in cutting speed would have little influence on the values of force and horsepower per cubic inch per minute but would affect the net power directly.

Comparing the force values of these two cutters for a depth of cut of 0.075 in. and a feed of 0.009 in. per tooth, it is seen, from Table 1, that, when changing from 25 to 45 deg helix, axial force was increased from 155 to 195 lb, or 25.8 per cent; tangential force was reduced from 800 to 495 lb, a decrease of 58 per cent; normal force was increased from 165 to 180 lb, or 9 per cent; and horsepower per cubic inch per minute was reduced from 1.715 to 1.62, or 5.2 per cent. These results show in general that the 45-deg helix angle cutter is more efficient than the 25-deg cutter when considering the tangential force and power. These percentages are changed considerably when the cutters are compared for a heavy cut having a depth of 0.200 in. and a feed of 0.0165 in. per tooth. For this condition, axial force was increased from 440 to 670 lb, or 52 per cent, while tangential force was decreased from 2650 to 1900 lb, or 28 per cent. Normal force was increased 48.5 per cent from a —700 to a —360 lb, and the power per cubic inch per minute required was increased from 1.36 to 1.39 hp, or 2 per cent. It would appear from these data that the 25-deg helix cutter was best for heavy cuts, while the 45-deg helix cutter was superior when taking light cuts. These same conclusions may be reached when analyzing the exponents of the formulas for the 25-deg and 45-deg cutters, as listed in Table 1.

Summary

The net horsepower per cubic inch per minute was found to vary directly as $Hp_n = K/d^x f^y$. Specific formulas for each cutter are given in Table 1. It is seen that the horsepower per cubic inch per minute changes with a change in rake angle. This change, however, produces a change in the constant K as well as the exponents of d and f. If the rake angle is reduced from 20 to 5 deg, the constant increases directly from 0.469 to 0.591; the feed exponent increases directly from 0.175 to 0.216; and the depth exponent decreases directly from 0.175 to 0.042. It is interesting to note that, if this same direct decrease of depth exponent is extrapolated, it would become 0 for 0-deg rake angle, thus indicating that, for a cutter with 0-deg rake, the depth would have no influence upon the horse power per cubic inch per minute. The total horsepower, then, would increase directly with the depth. These horsepower formulas indicate, by the value of their exponents, that 5-deg rake angle cutters are most efficient when using large feeds, while the 20-deg rake cutter is most efficient when taking deep depths of cut.

Comparison of Results With Other Experiments

It is the intention of the authors at some time to present a comprehensive comparison of all data obtained by them on the subject of milling energy as a function of feed, depth, and tool shape. This study represents a continuation of work reported on in two previous papers[5] (see footnote 4 on p. 545). Work along this line also has been done by other authors, and a thorough analysis of results and a direct comparison is desirable.

Conclusions

The following brief statements may indicate the conclusions arrived at from the results presented in this paper. It must be borne in mind that the material cut consisted of a hot-rolled annealed S.A.E. 3150 steel which was being cut by slab milling cutters of various shapes, as listed in Table 1, when a sulphurized-base cutting fluid was used. For each cutter, feed and depth of cut were varied. Speed of cutting, however, remained constant, except for the 8-tooth, 45-deg helix angle cutter for which the speed was increased in order that the feed per tooth would remain the same as for the 12-tooth cutters.

(1) Tangential and axial forces, net input power at the cutter, and net horsepower per cubic inch per minute all can be represented, as a function of depth of cut and feed, by mathematical equations, inasmuch as the data gave straight lines when plotted on double logarithmic paper (see Table 1).

(2) Normal force was found not to vary directly with depth of cut and feed but to give curves starting at a minimum, rising to a maximum, and falling off to a minimum, in many cases negative, as the feed was increased for a constant value of depth or as depth was increased for a constant value of feed. For most of the heavy cuts the normal force becomes negative, that is, the cutter tends to lift the work from its support (See Figs. 4, 10, 16, 22, and 29).

(3) The cutter having the larger rake angle of 20 deg was found to be more efficient from a power and force point of view when taking both light and heavy cuts than the cutter having less rake but operating under the same conditions. The large-rake-angle cutter appeared to be more efficient than the small-rake-angle cutter for the heavier cuts than for the lighter cuts.

(4) When cutting under similar conditions tangential force was found to be reduced in almost direct proportion as the rake angle of the cutter was increased from 5 to 20 deg (see Fig. 32).

(5) Normal force is reduced also in almost direct proportion with an increase in rake angle.

(6) Axial force remains practically constant while the rake angle is increased from 5 to 10 deg but is reduced rapidly in direct proportion for a further increase in rake angle.

(7) Horsepower per cubic inch per minute is reduced in direct proportion to an increase in rake angle from 5 to 15 deg and is further reduced but at a slower rate as the rake is increased further to 20 deg.

(8) Change in force or power, as feed and depth of cut are varied, is found to produce a change in the constant and the exponents of depth and feed in the equations representing each factor.

(9) For a light cut of 0.075 in. depth and 0.009 in. feed per tooth, and a 15-deg rake angle, a change in helix angle from 25 to 45 deg produced an increase in axial force of 25.8 per cent, a decrease in tangential force of 58 per cent, an increase in normal force of 9 per cent, and a reduction in horsepower per cubic inch per minute of 5.2 per cent (See Figs. 8 to 13 and 26 to 31).

(10) For this increase in helix, a heavy cut having 0.200 in. depth and 0.0165 in. feed per tooth, axial force was increased 5.2 per cent, tangential force decreased 28 per cent, normal force increased 48 per cent, and horsepower per cubic inch per minute was increased 2 per cent.

Acknowledgments

Grateful acknowledgment is made to the executive board of the Horace H. Rackham School of Graduate Studies of the University of Michigan for funds with which assistants were employed to conduct the tests.

Goddard & Goddard Co. generously furnished the cutters, specially made for this investigation. The milling machine was furnished by Cincinnati Milling Machine & Cincinnati Grinders, Inc.

The design and construction of the dynamometer is due in no small part to C. E. Kraus, then instructor in the department of metal processing, and Malcolm Loveland, and D. D. Stegenga, assistants in the department of metal processing.

The tests were run almost entirely by K. B. Kaiser.

Determining the Tool-Life Cutting-Speed Relationship by Facing Cuts

By C. E. KRAUS[1] and R. R. WEDDELL,[2] Rockford, Ill.

This paper presents the results of a series of tests made to obtain the tool-life cutting-speed relationship for a variety of high-speed steels when cutting a dense nickel cast iron. The cuts were taken on the face of a segmented disk, and the conditions of the cut were selected to approximate the conditions of a face-mill cut. A mathematical analysis is included and the data are put in the form of the equation $VT^N = C$. This method of testing has several advantages: The tests take less time, the point of failure is more definite when cutting cast iron, and material variations have practically no effect on the comparative accuracy of the data. Twenty-three steels were tested, ranging from 18-4-1 to supercobalt, including several molybdenum steels. A number of conclusions are given comparing tool life as affected by analysis. Also, the data are shown plotted on log-log coordinates. All tests were run dry under identical conditions.

The tool-life cutting-speed relationship conforms to the mathematical equation $VT^N = C$, in which V is the cutting speed, fpm; T is the tool life, N is an exponent and is the slope of the curve plotted over values of T on log-log paper; and C is a constant dependent on the conditions of the cut. This equation may be obtained directly for a given set of conditions by a number of tests at various cutting speeds, on a cylindrical test bar.

When the primary object of such tests is to compare materials, accurate comparative data may be obtained with relatively small test bars. When the object of the tests is to compare tool materials, or the effect of their heat-treatment, or of cutting fluids on tool life, it is essential to have a material to be cut which is uniform for all tests. If it were not, certain control tests must be repeated at frequent intervals and corrections made to the data. A cylindrical log presents variations in two directions—from the outside to the center and from end to end. If the log is steel, carefully forged, normalized, and annealed, these variations are small. If it be of cast iron, the variations can easily be so large as to make reasonably accurate results almost impossible.

This paper presents a method of testing and an analysis, which, it is believed, makes possible quite accurate results on cast iron. Data from one series of tests are included to illustrate the method.

The use of face cuts for determining tool-life data has been tried many times. With this method a tool is started at the center of a disk of material revolving at a constant speed and fed radially outward until it fails. The cutting speed for a given point is a function of both the speed in revolutions per minute and the radius, and the radius at the point of failure is a function of both the speed in revolutions per minute and the tool material for any given test conditions. If various tool materials are tested at a given number of revolutions per minute, the values for the radius at failure will be found to vary somewhat and it is usually assumed that the larger the radius the better the tool material. That this is not necessarily true is soon discovered when trying to correlate the results of such tests with the results of tests on cylindrical cuts, or results obtained from practice. Also, a different order for some of the tool materials is obtained if the speed of the test disks is changed.

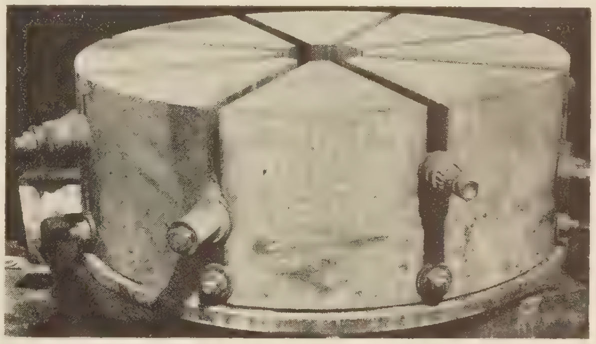

Fig. 1 Assembled Test Block of Nickel Cast Iron

This seeming lack of correlation is due to the change in slope of the cutting-speed-tool-life curves for the different tool materials, and therefore the actual test data obtained from face-cut tests are of little value unless put in the form of the $VT^N = C$ equation.

Description of Test Block and Procedure

The specific series of tests presented in this paper were designed to compare various types of high-speed steels when cutting cast iron under conditions as approximating those encountered in face milling.

It is planned to use tool-life tests of this type as a basis of selection of high-speed steels and as a check on the heat-treatment given cutter blades. Eventually, other materials may be tested since it is well-known that the various high-speed steels react differently on different materials.

The machine used was a 56-in. Bullard vertical turret lathe, belt-driven by a 20-hp a-c motor. Two faces of the turret were used, one for the cleanup tool and one for the test tool. The cleanup tool was used at slow speed and ground by hand as often as necessary to keep it reasonably sharp.

Fig. 1 shows the assembled test block which was mounted on the table of the Bullard lathe. It is built up of eight sectors of a dense wear-resisting cast iron containing about 60 per cent steel

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1937, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

in the mix. The analysis of the iron was 2.90 to 3.10 per cent total carbon, 1.30 to 1.40 per cent silicon, 0.80 to 0.90 per cent manganese, and 1.20 to 1.30 per cent nickel. Transverse tests of this iron with a 12-in. span and a 1.2-in. diameter bar give a breaking load of 5150 lb with a deflection of 0.117 in. The tensile strength is 54,000 lb per sq in.; the modulus of elasticity is 19,700,000 lb per sq in., and the average Brinell hardness is 269. A 1-in. space between sections gives an interrupted cut, roughly equivalent to face milling.

In order to speed up the tests and conserve material the chip was turned over at right angles and a feed of 0.131 in. per revolution was used with depth of cut 0.025 in. The effect on the tool is the same as if the feed was 0.025 in. and the depth was 0.131 in.

High-speed steel fails when cutting cast iron by abrasion on the clearance face. As the abrasion continues, a flat is worn which, by rubbing against the work, develops considerable heat, and which in turn increases the rate of abrasion. When the temperature at the point of rubbing exceeds the draw point of the steel the abrasion is accelerated and complete failure soon follows. At the time this acceleration begins red-hot particles are visible on the work behind the tool. For relatively light cuts there is no point where it can be definitely said failure has occurred, as is possible when cutting steel or any material producing a crater type of failure. The only requirement for comparative tests is that the point, at which failure is assumed to occur, must be for the same amount of wear for all tools. For small depths or finishing cuts, a common method is to use a trailer tool set for a given wear and stop the test when it begins to cut. For these tests it was thought sufficiently accurate to watch carefully for the first signs of fire, since it was soon noticed that this point could be obtained after a little experience with an accuracy of less than one revolution of the work. However, it should be remembered that the tool wear at this point is rather more than is usually allowed in practice, and the results are, therefore, somewhat high. Another problem involved in testing cast iron is the change in chip area resulting from this tool wear. When making tests on a cylindrical bar the only accurate procedure is to clean up the full length of the cut at slow speed with, preferably, a carbide tool after each test. When this is done the depth of the chip gradually becomes less as the wear continues. If the feed is light compared to the depth, as is usual in face milling, most wear occurs in the direction of feed, and the chip area is little affected. When the feed is large compared to the depth, a slightly saucer-shaped surface is left when taking a face cut and the area of the chip is greatly affected if a plane surface was present at the start of the tests. In order to avoid this trouble it is only necessary to leave the saucer-shaped surface of the preceding cut, and no subsequent cleanup cut need be taken over the surface cut by the test tool. The procedure followed in these tests was to clean up only enough inside of the point where fire was first visible, so as to insure the absence of hard tool particles which might be embedded in the work at that point.

The accuracy of these tests is considered to be good. The maximum experimental error of the individual points was about 5 per cent and the average error was about 2 per cent. The values of the constants are correct within about 2 per cent and the values of the exponents will vary somewhat more, especially as the exponent gets smaller.

Tool Materials

Table 1 lists the tool materials tested divided into the following groups: Nos. 1 to 7, inclusive, are 18-4-1 steels in which it should be noted that steel No. 7 has higher than normal vanadium content. No. 8 is a permanent mold casting of high-speed steel the exact content of which is unknown. Nos. 9 to 12, inclusive, are 18-4-2 varieties. Nos. 13 and 14 are molybdenum

TABLE 1 ANALYSIS OF TOOL MATERIALS AND TEST RESULTS

Tool material no.	Analysis, per cent W	Cr	V	Mo	Co	Rockwell C hardness	Cutting-speed-tool-life equation
1	18.35	3.79	1.12	..	...	63.0	$VT^{1/11.75} = 96$
2	18.00	4.00	1.00	..	...	62.0	$VT^{1/11} = 96$
3	18.00	4.00	1.00	..	...	64.0	$VT^{1/8.14} = 110$
4	18.00	4.00	1.00	..	...	64.5	$VT^{1/8.84} = 110$
5	18.00	4.00	1.00	..	...	62.5	$VT^{1/8.41} = 110$
6	18.00	4.00	1.00	..	...	63.5	$VT^{1/8.7} = 110$
7	18.00	4.00	1.40	..	...	63.0	$VT^{1/12.4} = 102$
8	...	..	..	..	...	64.5	$VT^{1/8.82} = 110$
9	18.00	4.00	2.00	0.50	...	62.5	$VT^{1/9.34} = 110$
10	18.00	4.00	2.00	0.65	...	64.0	$VT^{1/9.0} = 112$
11	18.00	4.00	1.85	0.90	...	65.5	$VT^{1/13.3} = 111$
12	...	..	..	..	...	65.0	$VT^{1/13.5} = 110$
13	1.50	4.00	1.25	8.00	...	65.5	$VT^{1/8.7} = 110$
14	2.50	4.00	1.35	8.00	...	65.0	$VT^{1/8.88} = 115$
15	1.50	4.00	1.25	8.00	5.00	65.0	$VT^{1/8.24} = 118$
16	12.50	4.00	2.10	..	2.75	62.0	$VT^{1/8.15} = 107.5$
17	14.00	4.00	2.00	0.50	5.00	64.5	$VT^{1/7.06} = 130$
18	18.00	4.00	1.00	..	3.16	66.0	$VT^{1/13.5} = 113$
19	18.00	4.00	1.00	0.50	5.00	63.0	$VT^{1/8.45} = 123$
20	17.25	4.00	1.00	0.50	4.50	61.5	$VT^{1/11} = 123$
21	18.50	4.50	1.75	1.00	9.00	62.5	$VT^{1/8.41} = 107$
22	20.50	4.50	1.30	0.60	12.25	65.5	$VT^{1/12.4} = 115$
23	18.00	4.50	1.40	0.70	10.00	63.5	$VT^{1/12.1} = 121$

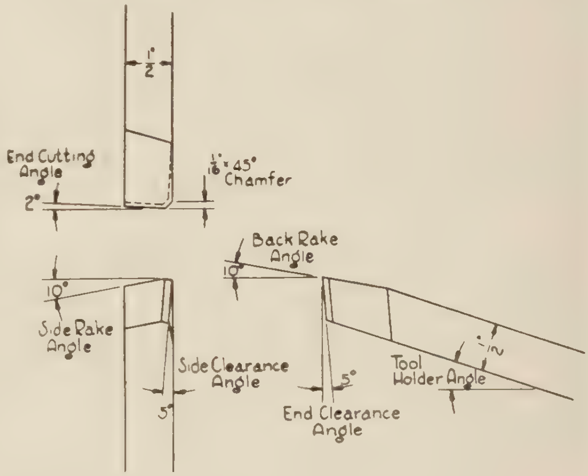

FIG. 2 TOOL ANGLES

high-speed steels. No. 15 is a cobalt molybdenum steel and the remainder contain varied amounts of cobalt. The analysis given is in most cases the type analysis.

Fig. 2 shows the tool angles used as standard for all tests, selected to duplicate the angles commonly found on face mills designed for cast iron. These angles may not be the most efficient for the material cut, but no effort was made to determine the best angles in this series of tests.

The test tools were very carefully ground on a surface grinder, cutting dry. No temper color was allowed to show, and after failure the tools were ground off on the end about $^1/_{16}$ in. by hand before regrinding in the surface grinder to insure unharmed steel in the cutting edge. The tools were not touched with a hone as it is felt that honing is rarely practical with milling cutters, although it is well-known that careful honing will increase tool life.

Discussion of Results

The data obtained from tests using the face-cut method are in the form of (1) values of R, revolutions per minute of the test block, and (2) corresponding values of r, the radius at the selected point of failure of the tool. These data are plotted on log-log paper, as shown in Fig. 3, in order to determine the slope of the resulting straight line.

For the tool shown, seven points were run and the data are quite consistent. The number of points run per tool varied from five to eleven in one case and averaged about seven. When

the data were erratic additional points were run until a reasonably accurate curve resulted. The slope of the curve on log-log paper plotted over the radius at failure gives the value of $(1 + N)/(1 - N)$. The corresponding value of N is shown in Fig. 4; for the tool shown $(1 + N)/(1 - N)$ equals 1.185 and N equals $^1/_{11.75}$. Substituting this value of N and a representative value of R and r from the curve it is but a few minutes work with a slide rule to determine the value of C, which in this case is 96.

The resulting cutting-speed-tool-life equations are listed in Table 1 and are shown graphically in Figs. 5, 6, 7 and 8, inclusive, on log-log paper. It will be noted that the values of the constant C vary from 96 to 130, and the values of the exponent N vary from $^1/_{70.6}$ to $^1/_{13.5}$.

Since the value of the constant is the cutting speed for a tool-life of 1 min, it is apparent that a high constant indicates a high resistance to abrasion at high temperature, and is therefore related to the red hardness. The value of the exponent is the

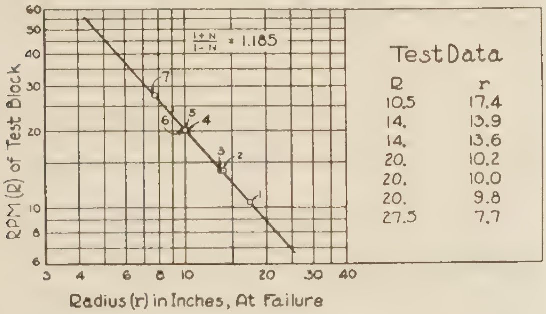

FIG. 3 FACE-CUT TOOL-LIFE DATA PLOTTED ON LOG-LOG COORDINATES TO DETERMINE THE VALUE OF $(1 + N)/(1 - N)$. TOOL MATERIAL NO. 1 USED IN TESTS

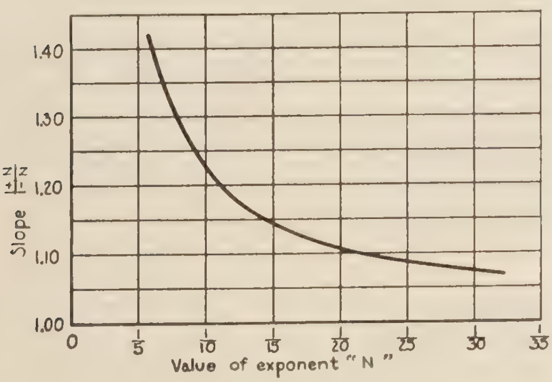

FIG. 4 RELATION BETWEEN SLOPE $(1 + N)/(1 - N)$ AND EXPONENT N

slope of the curve plotted on log-log paper. A flat curve having a low value of the exponent indicates greater wear resistance under more normal cutting conditions. Therefore, it follows that a high constant does not necessarily mean the best steel. The best combination is a high constant and a low exponent, but for a relatively long tool-life the low exponent has more effect on cutting speed than the high constant.

Apparently a number of factors influence these values, and it is not the intention of the authors to go thoroughly into this phase of the problem in this paper as the subject is too involved. However, the various types of steels are separately discussed in the following paragraphs and some comments are made on the range of these values as influenced by the analysis.

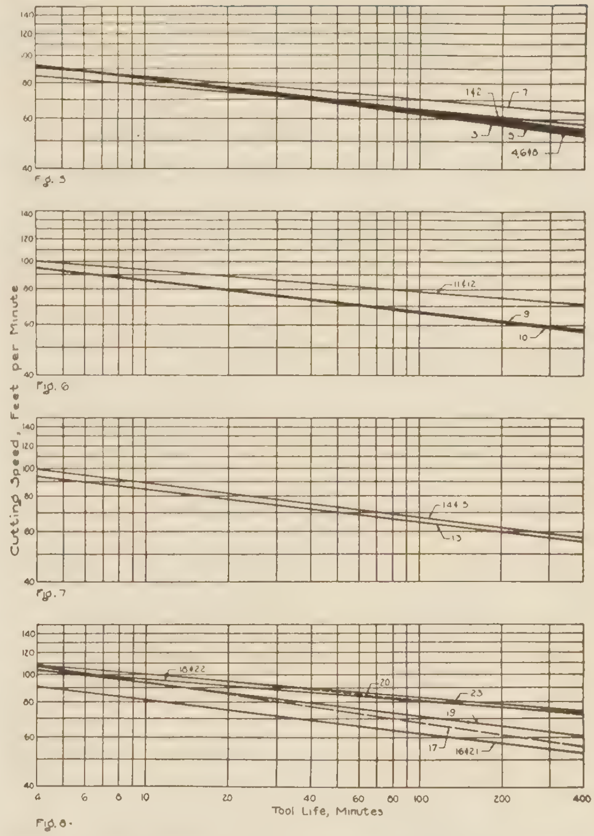

FIG. 5 CUTTING-SPEED-TOOL-LIFE CURVES FOR 18-4-1 HIGH-SPEED STEELS

FIG. 6 CUTTING-SPEED-TOOL-LIFE CURVES FOR 18-4-2 HIGH-SPEED STEELS

FIG. 7 CUTTING-SPEED-TOOL-LIFE CURVES FOR MOLYBDENUM-BEARING HIGH-SPEED STEELS

FIG. 8 CUTTING-SPEED TOOL-LIFE CURVES FOR COBALT-BEARING HIGH-SPEED STEELS

Fig. 5 shows the curves for the 18-4-1 analysis steels. Steels Nos. 1 and 2 each have a constant of 96 and a slope of about $^1/_{11}$. Steels Nos. 3 to 6, inclusive, fall into another group having a constant of 110 and slopes of about $^1/_{8.5}$. Steel No. 7 should more properly be plotted on Fig. 6 since it has considerably more than 1 per cent vanadium and its curve will be found to fall in the middle of the 18-4-2 group of curves. It will be noted that there is more uniformity in the 18-4-1 group than in any other, which may be attributed to more nearly standardized heat-treatment commonly used. Steel No. 8 is rather interesting. It is made by melting cuttings of high-speed steel, adding alloying elements as may be required, and casting in metal molds. The resulting casting may or may not be heat-treated, but in many cases better results are obtained in the "as-cast" condition. For the test conditions here reported this material gives the same results as the ordinary 18-4-1 steel.

Fig. 6 shows four 18-4-2 steels which fall apparently into two groups. The constants however are alike and the same as the largest 18-4-1 group. Steels Nos. 9 and 10 have exponents of about $^1/_9$ while Nos. 11 and 12 have exponents of about $^1/_{13.5}$. The average slope is flatter than for the 18-4-1 steels, and the addition of the extra vanadium appears very much worth while.

Fig. 7 shows curves for three molybdenum steels. Nos. 13 and 14 contain 8 per cent molybdenum, and steel No. 15 contains, in addition, about 5 per cent cobalt. All three have slopes comparable to the 18-4-1 group, but have a somewhat higher average constant.

The greatest variety, both in results and analysis, is found in the cobalt-steel group plotted in Fig. 8. This group can be subdivided as follows: Nos. 16 and 17 are of a 14-4-2 type with 2.75 and 5 per cent cobalt, respectively. Nos. 18, 19, and 20 are of an 18-4-1 type with about 5 per cent cobalt added. Nos. 21 and 22 contain 9 and 12.25 per cent cobalt, respectively, and more than 1 per cent vanadium. It is immediately apparent that

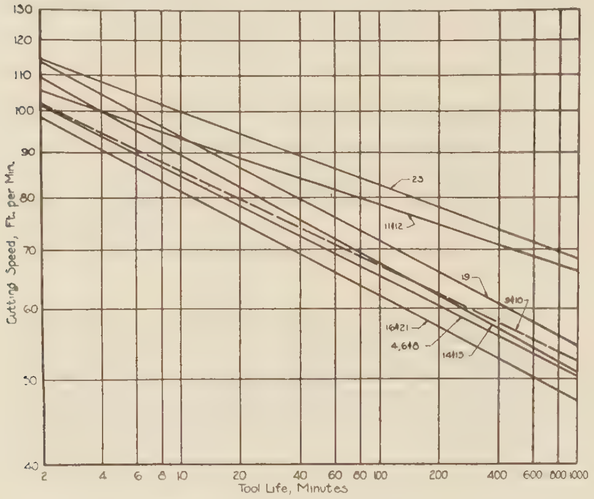

Fig. 9 Cutting-Speed-Tool-Life Curves for Various Types of High-Speed Steel Plotted on Modified Log-Log Coordinates. Chip Size Was 0.131 × 0.025 In. All Tests Were Made on Nickel Cast Iron, Cutting Dry

the results will not correlate with the analysis. The authors on other occasions have many times run tests on some of these steels with the same results, but of a different order. For example, steel No. 21, which is one of the two lowest in these tests, is consistently one of the best on tough steels. In general, too, tool bits of the same brand will show less uniformity from bit to bit than other classes of high-speed steel, although any given bit will duplicate results test after test with better than average accuracy. The average value of the exponent for the group is lower than that for the 18-4-1 group and about the same as that for the 18-4-2 group. The average value of the constant is considerably higher than the other groups.

In order to better compare the different types of high-speed steels tested, a few selected curves have been plotted on a modified log-log scale in Fig. 9. The curve labeled 4, 6, and 8 represents the majority of the 18-4-1 steels and also represents No. 13 which is a molybdenum steel. The other two molybdenum steels Nos. 14 and 15, are a little higher on the scale. The dashed line, for steels Nos. 9 and 10, is in about the same place and represents two of the 18-4-2 steels. The curve for steels Nos. 11 and 12 also represents 18-4-2 steels. The lowest and the highest curves shown are for cobalt steels. Fig. 9 shows the very large variation in results obtained from these steels. At a cutting speed of 70 fpm, the tool life may range from 35 to 750 min and for a tool life of 600 min the cutting speed may vary from 50 to 72 fpm.

Conclusions

A number of conclusions may be drawn from the work reported and are listed as follows. The conclusions drawn are rather general, but will apply strictly only for the conditions of the tests reported:

1 Face-cut tests for determining tool-life data appear to be very advantageous when cutting cast iron.

2 Such tests have proved fast and accurate.

3 Data obtained by the face-cut method may easily be put into the usual fotm of equation $VT^N = C$.

4 When cutting a chip 0.131 × 0.025 in. with an interrupted cut in a dense nickel cast iron, cutting dry, the following comparisons of the performance of various types of high-speed steels may be made: (*a*) As a group, the 18-4-1 steels gave the most consistent results. (*b*) Increasing the vanadium content lowers the exponent, increasing the tool-life in the usual range of cutting speeds. (*c*) The molybdenum steels are slightly better than the 18-4-1 and not as good as the 18-4-2 steels. (*d*) The least consistent results were obtained with the cobalt steels; some were inferior to the 18-4-1 group. The average cobalt steel has a higher constant than other types.

Analyzing Variable Loads in Steam- and Diesel-Power Plants

By GLENN C. BOYER,[1] KANSAS CITY, MO.

This paper presents a rapid and accurate method for determining the total input to a machine supplying a predetermined variable output. While the method is adaptable to any variable load condition, it is concerned primarily with the adaptation of the method to Diesel and steam electric-generating stations.

Essentially the method consists in using a load-duration curve of the load studied together with an input-output curve for the Diesel engine or steam equipment considered for supplying the load. By the use of these two curves and mechanical projection, a duration curve of input is developed. The area under the load-duration curve is proportional to the total output of the plant for the period considered, and similarly the area under duration curve of the input is proportional to the total input.

Where plant load-duration curves can be prepared covering an entire year's operation, this method provides a means for predetermining the total amount of fuel necessary to generate this quantity of energy. The accuracy of the method is influenced by the accuracy of the load-duration curves available, and the errors involved in the input-output curve used.

A brief discussion is included which indicates the adaptability of the method to other uses than those to which the paper is primarily devoted.

ENGINEERING studies of steam- and Diesel-driven electric generating plants, by the organization with which the writer is associated, have shown the need of a rapid method for obtaining the average performance of a contemplated installation over a variable yearly load cycle. In speaking of the performance of a particular generating station we constantly refer to "Btu per kwhr generated" when discussing steam prime movers or "kwhr per gallon of fuel oil" for Diesel prime movers. After the station is in operation, these figures are readily obtainable from the plant performance record. It is only when we attempt to predetermine this information for a proposed installation operating under known or assumed conditions that difficulties appear.

The difficulties arising in connection with the problem result from the fact that the load served by an electric-generating plant varies. In most instances that have come to the writer's attention, the maximum load carried by a plant ranges from 4 to 6 times the minimum load during the period of one year. Recent studies of a midwestern generating plant having a 60 per cent annual load factor, showed a ratio of 4 to 1 from maximum to minimum load. The load factor of the plant appears to bear no relation to the spread between maximum and minimum load on the plant.

In another instance the maximum load on a small generating plant was 1050 kw, and the minimum load 175 kw, or a ratio of 6 to 1 from maximum to minimum load. During this same year the plant produced a total of 3,854,400 kwhr and the annual load factor was 41.9 per cent. It was desired to make certain improvements in generating equipment in this plant operating with steam prime movers at 175 lb per sq in. gage pressure, and it became necessary, therefore, to estimate accurately the performance of contemplated equipment in order to make a comparison with present performance.

When discussing load variation in a generating station, the tendency has been to attempt to show load variation by means of daily load curves. Fig. 1 contains two daily load curves for this plant for the maximum and the minimum day during one year. While these load curves show what happened throughout the 24 hours of two specific days, they can never give a complete picture of the load variation which that plant was called upon to furnish throughout the year. In order to present the complete story of load variation, it will require not two but 365 daily load curves. Should the attempt be made to plot all 365 curves on Fig. 1 they would interlace to the point of being meaningless.

The first necessity, therefore, is a means for presenting the load variation for a total of 8760 hours, or one year, in such a manner that the data will be of value. The load-duration curve, Fig. 2 is the best-known means for presenting the variation in load for any period of time in a manner that lends itself to ready interpretation. As plotted in Fig. 2, any point on this curve shows the per cent of the total hours in the year when the load exceeded a given value. Thus, while the peak load on the plant was 1050 kw the load exceeded 575 kw 20 per cent of the time, and 450 kw 50 per cent of the time.

While reference has been made to the load-duration curve in various published literature of the major engineering societies, the paper by H. Alden Foster (1),[2] and the article by A. G. Christie (2), are the best that have come to the writer's attention. Since these papers present the duration curve in detail, particularly with regard to its development, that phase will not be dwelt upon here.

It should be pointed out in passing, however, that since the abscissa or horizontal axis of the duration curve, Fig. 2, represents time and the ordinate or vertical axis represents load, the area under the curve represents power output. This property of the curve will be referred to in greater detail later. The load-dura-

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1937, for publication at a later date. Discussion received after this date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Numbers in parentheses refer to Bibliography at end of paper.

tion curve can also be employed as a graphical representation of annual load factor since the ratio of the area under the duration curve to the area of the rectangle bounded by the X and Y axes, the abscissa for the maximum load on the plant and the ordinate for 100 per cent total hours in the year is annual load factor.

Application of Diesel Engines

In the consideration of an improvement program for the electric-generating plant previously referred to, it is desirable to know the kilowatt hours generated per gallon of fuel oil during a year's period if Diesel prime movers were used in place of steam turbines. The load-duration curve in Fig. 2 was derived from the operating data for this plant and will be employed in the analytical method presented in this paper. In addition to the load-duration curve the guaranteed performance data for the Diesel-engine unit or group of units under consideration is also necessary.

For this particular plant, one of the plans under consideration utilized two Diesel-engine generating units each having a full-load capacity of 870 kw. The operating schedule called for the second

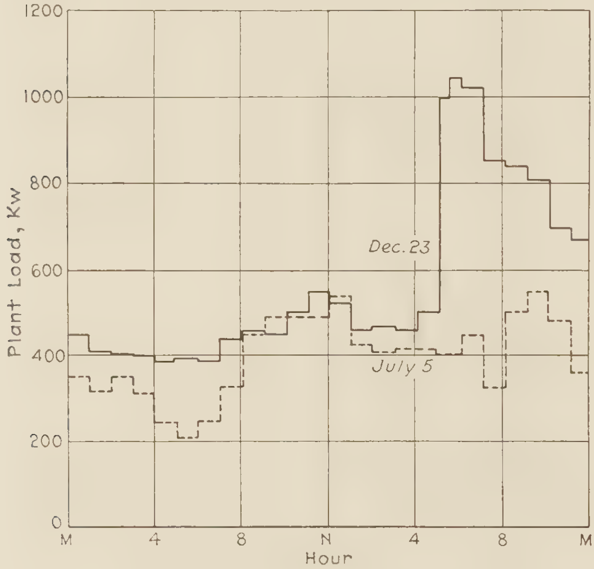

Fig. 1 Daily Load Curves

unit to be started when the plant load reached 750 kw and the plant load above this value was to be divided equally between the two machines. With this operating schedule and the fuel guarantees of the units under consideration, a curve of engine performance can be plotted with plant load in kilowatts as the ordinate and gallons of fuel oil per hour as the abscissa.

In the lower right-hand corner of Fig. 3 is reproduced the load-duration curve from Fig. 2, and in the lower left-hand corner of Fig. 3 is plotted the performance or input-output curve for the group of generating units considered. In plotting the load-duration curve the abscissa values increase in the right-hand direction from the origin, while in the case of the input-output curve the abscissa values increase in the left-hand direction from the origin. The values for the ordinates are common to both curves.

With the use of these two curves and elementary mechanical projection, a third or fuel time is developed in the upper right-hand corner in which the abscissa is per cent total hours and the ordinate is gallons of fuel oil per hour. The mechanics of the development of the fuel-time curve is shown by means of the dashed lines and arrows for locating one point on the curve. Fundamentally, this curve shows the per cent of time during which the rate of burning fuel oil exceeded a certain number of gallons per hour. Since the abscissa of this fuel-time curve is time and the ordinate rate of oil consumption, the area under the curve is proportional to the quantity of fuel oil consumed in generating the quantity of electrical energy represented by the area under the load-duration curve. This proportion is readily obtained. If, for example, one inch horizontally represents 20 per cent of the total hours in the year or 1752, and one inch vertically

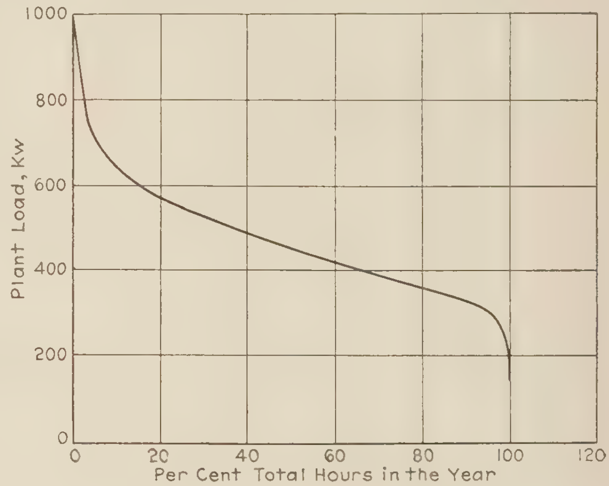

Fig. 2 Load-Duration Curve

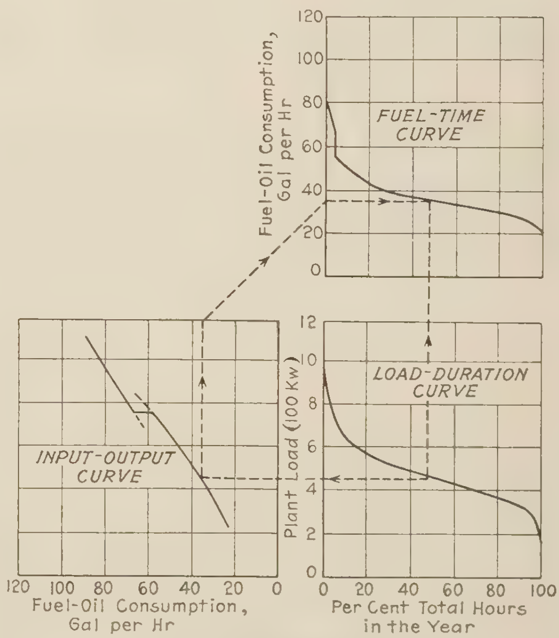

Fig. 3 Diesel-Engine Analysis

represents 10 gallons of fuel oil per hour, then one square inch of area under the fuel-time curve represents 17,520 gallons of fuel oil consumed.

For office use the three curves on Fig. 3 are plotted on a sheet of cross-section paper approximately 17 in. × 20 in. Since standard cross-section paper divided 10 × 10 to the inch is available in rolls 20 in. wide, no difficulty is experienced in obtaining sheets of practically any dimensions for developing the curves in Fig. 3.

Areas under the load-duration and fuel-time curves can be obtained either by the use of a planimeter or by counting squares. From curves developed in connection with this analysis for Diesel-engine-generating units, the following information was obtained.

TABLE 1 SUMMARY OF DIESEL-PLANT ANALYSIS

Total energy generated (actual), kwhr	3,854,400
Total energy generated (area under curve), kwhr	4,150,000
Per cent error	7.5
Total fuel oil consumed (area under curve), gal	351,000
Kilowatthours generated per gallon oil $= \frac{4,150,000}{351,000} =$	11.85
Btu per kwhr generated	11,800
Over-all plant efficiency, per cent	29

It is seen from Table 1 that an error of 7.5 per cent is introduced in development of and determination of the area under the load-duration curve. For analytical purposes, all calculations are based upon the areas found under the various curves, due to the fact that an error introduced into the load-duration curve is automatically reflected in the fuel-time curve. By using the values determined from the curves, the per cent error in the figure for kilowatthours generated per gallon of fuel oil is less than if the actual kilowatthours generated and the value for the total fuel oil consumed as determined from the fuel-time curve are employed. By using the latter combination of values a 7.5 per cent error is automatically introduced into the value for the quantity of fuel oil consumed, without any compensating error in the quantity for total generation in kilowatthours.

The input-output curves for contemplated improvements employing other sizes of generating units could be plotted on the lower left-hand portion of Fig. 3, and their respective fuel-time curves developed. These would enable the designing engineer to readily select the most economical combination of units from the fuel-economy standpoint.

Application to Steam Turbines

Although somewhat more involved, the same procedure can be used for the study of contemplated steam-plant improvements. With the load-duration curve in Fig. 2 and the heat-balance calculations for a proposed steam cycle utilizing a 2000-kw turbine operating at 400 lb per sq in. gage, 750 F total temperature and a 28 in. vacuum, the basic information is available for starting Fig. 4. As in Fig. 3, the load-duration curve, Fig. 2, is plotted in the lower right-hand corner and the input-output curve for the steam cycle is plotted in the lower left-hand corner. The steam-time curve is then developed by means of mechanical projection immediately above the load-duration curve. It is necessary to consider the efficiency of the boiler to know the quantity of fuel which would be burned. By plotting an input-output curve for the boiler to the immediate left of the steam-time curve and above the input-output curve of the steam cycle and by mechanical projection, a fuel-time curve is developed. The boiler input-output is based upon the use of bituminous coal running 9000 Btu per lb and a boiler with 4000 sq ft of heating surface exclusive of the superheater.

The three curves to the right of the major vertical axis consisting of the load-duration curve, based upon actual operating data, and the steam-time and fuel-time curves which were developed mechanically, afford a complete picture of the operation of the proposed plant under known load conditions. From them can be obtained the following information:

(*a*) Total kwhr generated as well as maximum and minimum station loads in kw.

(*b*) Total steam produced by the boilers as well as maximum and minimum rates of steam production.

(*c*) Total coal burned as well as maximum and minimum rates of burning.

From the curves developed in Fig. 4, Table 2 was formulated.

TABLE 2 SUMMARY OF STEAM-PLANT ANALYSIS

Total energy generated (actual), kwhr	3,854,000
Total energy generated (area under curve), kwhr	4,150,000
Per cent error	7.5
Total steam requiréd (area under curve), lb	55,950,000
Total coal 9000 Btu per lb (area under curve)	10,510,000
Pounds steam per kwhr generated	13.5
Pounds coal per kwhr generated	2.54
Btu per kwhr generated	22,800
Plant thermal efficiency, per cent	15
Rate of steam production, maximum, lb per hr	11,750
Rate of steam production, minimum, lb per hr	4,000
Rate of coal consumption, maximum, lb per hr	2,100
Rate of coal consumption, minimum, lb per hr	875

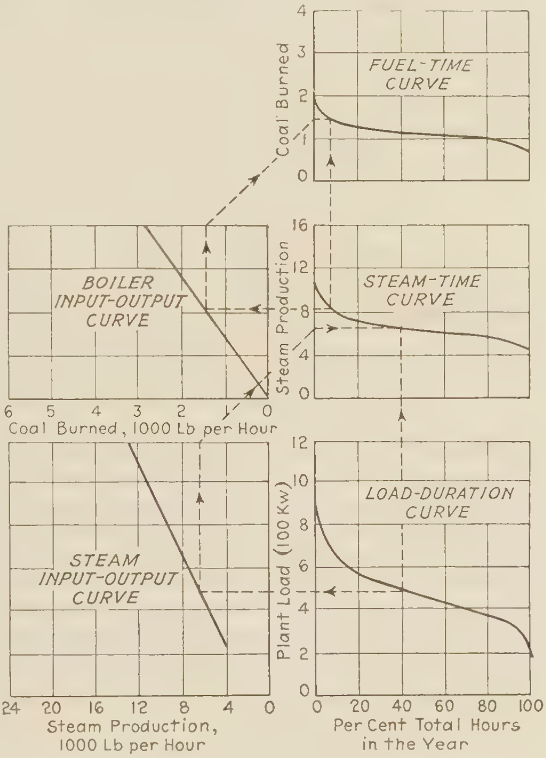

Fig. 4 Steam-Turbine Analysis

When applied to existing equipment in a small steam-generating station the results in terms of pounds of coal per kilowatthour generated obtained by the foregoing analytical method have checked within a range of 7 to 10 per cent of actual coal consumption.

Other Applications

The analytical method outlined in this paper may be applied to any power study where it is desired to obtain either the input, output, or efficiency curve of any power-generating or consuming unit when data for two of the curves are available. For example, this method has been used by the author for determining the electrical input to a motor-driven pump for waterworks service, when the variations in rate of flow and the pumping head were known.

It is rapid, reasonably accurate, and permits direct comparison of any number of proposed power schemes for supplying any variable load.

Acknowledgment

The author wishes to acknowledge the many comments and criticisms from those to whom this method of analysis has been

presented, and particularly those of Eric Therkelsen, professor of mechanical engineering at Montana State College; George C. Shadd, dean, Earl D. Hay, head of department of mechanical engineering, and D. C. Jackson, Jr., formerly professor of electrical engineering, University of Kansas; Leon B. Reynolds, professor of hydraulic and sanitary engineering at Stanford University, and those professional associates in his own company who have helped in applying the acid test of workability to this analytical method.

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Supplementary Data

FOLLOWING the preparation of this paper for presentation before the Semi-Annual Meeting of the A.S.M.E. at Dallas in June, 1936, the Oil and Gas Power Division reviewed it and suggested inclusion of several points not specifically mentioned in the original draft of the paper. These points together with the author's discussion of them follow:

1 *Reduction in Diesel-Engine Capacity Due to Altitude.*

The only effect this would have would be to shift the location of the input-output curve in Fig. 3 to the right when the unit was operating at elevations greater than 1500 ft above sea level. The amount of shift in this curve would be dependent upon the specific conditions governing a particular case. Needless to say, it is impossible to cover all of the variations which can occur in the development of an input-output curve for a particular installation. The necessary adjustments and variations must be made by the one using this analytical method in light of the guaranteed performance of the equipment in question in a specific location.

2 *Loss of Capacity Due to Summer Temperatures.*

All Diesel-engine guarantees are subject to a tolerance of 5 per cent as a standard practice, with the air-intake temperature not exceeding 90 F. Air temperatures in excess of this figure will result in additional losses in the unit. Compensation for higher temperatures during summer months involves so many variables and unknown factors that the author has found it advisable to ignore it in this analytical method. Ignoring it adds an error in the analysis. However, the error involved in neglecting temperature conditions above 90 deg was felt to be of such slight consequence that it could be done safely.

3 *Loss in Diesel Efficiency After 2 or 3 Years' Operation.*

Here again is a question that should be viewed in the light of the use an engineer makes of this analytical method. If, as is the case with most of the author's activities, the method is used for comparative purposes, all equipment must be compared on a common basis. Consequently, the comparison may be made without reference to future conditions. After an analysis concerning a Diesel engine has been made for one condition, an allowance can readily be made for loss in efficiency, if in the opinion of the engineer making the analysis such an allowance is necessary in the light of the problem in hand.

4 *No Allowance Made for Plant Auxiliaries.*

Allowance was made for auxiliaries driven by steam in the case of the steam-power analysis. Such an allowance was not made for electrically operated pumps in the case of the Diesel plant. In either instance, the curves consider the gross electrical output of the plant.

5 *No Consideration Given to Occasional Outage of Units.*

This is correct. In the interest of simplicity in presenting an analytical method, many possible combinations of units were not considered. If the units in a particular plant are all of the same size this question of outage of a unit will have no bearing on the analysis. If several sizes of units are in operation or under consideration, analyses will have to be made for the various possible combinations of units and some "horse sense" used in interpreting the results obtained.

6 *Weight per Gallon of Fuel Oil Used.*

The fuel considered for this analysis was Standard Oil Company special Diesel fuel ranging in gravity from 28 to 30 API, weight 7.29 lb per gal, Btu per gal 140,000.

7 *Steam Analysis—Plants With More Than One Turbine and One Boiler.*

This is only an extension of the method of analysis already covered in the paper. It has been the intention of the author to attempt to explain an analytical method and illustrate it with as few examples as possible in order to make the method clear to other engineers who might want to use it. Consequently the writer purposely omitted many analyses which might properly have been included in this paper. Likewise, much discussion of points not bearing directly on the analytical method was omitted for the sake of brevity and clarity. The author recognizes that this analytical method is only one tool for the engineer's work kit. To cover all of the elements having a bearing on the analysis of a problem involving variable loads was obviously impossible and the question foremost in the author's mind in preparing the article was just where to stop.

Experimental Investigation of the Influence of Tube Arrangement on Convection Heat Transfer and Flow Resistance in Cross Flow of Gases Over Tube Banks

By ORVILLE L. PIERSON,[1] NEW YORK, N. Y.

New measurements of the convection heat-transfer rate between gases and tube banks, with corresponding pressure drop for transverse flow have been made as part of a research program[2] of the Babcock & Wilcox Company to determine the effect of varying the spacing of tubes of identical size. This portion of the program was conducted by the Babcock & Wilcox Company in the mechanical-engineering laboratories of Rensselaer Polytechnic Institute. Thirty-eight arrangements of tubes were tested, differing in center-to-center spacing in the direction of flow and transversely, and these are shown in Fig. 1. The spacing, for both in-line and staggered tube rows varied from the closest practicable to three tube diameters. Each bank with two exceptions comprised ninety tubes, arranged in ten rows transverse to the air stream, each row containing nine tubes. These tubes were 0.31 in. in diameter and uniformly heated electrically throughout the $9^1/_8$-in. length from which heat was transferred to a cool air stream. Heat input to the bank as indicated by wattmeters in the supply line was held constant at 72,000 Btu per hr while the air mass flow was varied. One exception included two tube banks in which the depth of the bank was varied from ten rows to one row with the same heat input per tube. In the other exception the heat input was reduced to 18,000 Btu per hr. Incidentally, the pressure drop was also determined for zero heat input.

Heat-transfer rates are reported in terms of the relation of the Nusselt number to the Reynolds number; and the pressure drop is reported in terms of the relation of the Fanning-equation friction factor to the Reynolds number. The Reynolds number was varied from 2000 to 40,000 to include the range of commercial practice, and in view of the small scale of the apparatus, this was accomplished by the use of air under pressure in a closed chamber.

These tests have demonstrated that both convection heat transfer and flow resistance of tube banks vary markedly with changes in tube arrangements, no simple statement of the variation being possible. It was also found that the rate of change of these characteristics with changes in the Reynolds number was regularly related to the tube arrangement. The air-boundary convection conductance was found to increase approximately as the two-thirds power of the Reynolds number. The actual exponent ranges from about 0.55 to 0.80, varying in some few cases with Reynolds numbers as well as with the tube arrangement. Friction factors were also found to vary with the Reynolds number. The relation ranges from the 0.2 power to about the −0.3 power when friction factors are expressed as exponential functions of the Reynolds number. The relation between the friction factor and the tube spacing is not the same as that for conductance of the gas boundary, and there is no consistent relation between pressure drop and heat-transfer rate.

Test Method

The extended testing of a large number of tube arrangements was made economically possible by the use of these small-scale model banks. Rational deductions indicate model tests should yield full-scale results for both convection transfer and flow resistance with gaseous fluids if tests are conducted at full-scale Reynolds numbers. The Reynolds number may be written

$$Re = \frac{D\rho V}{\mu} \qquad [1]$$

where D = characteristic dimension of tube bank, such as the tube diameter; ρ = specific weight of fluid; V = fluid velocity; and μ = absolute viscosity of fluid.

It is apparent that a decrease in D may be compensated in several ways. One is to increase V, but it soon approaches values where compressibility effects introduce uncertainties. The viscosity may be varied by changing fluids, but gases offer little choice in this regard and the use of totally different fluids greatly complicates the situation. The method of maintaining full-scale Reynolds numbers used in the present case increased the specific-weight term by compressing the gas, using the same gas and velo-

[1] Analytical Engineer, Babcock & Wilcox Company. Dr. Pierson was graduated with a B.S. degree from the University of Nebraska, and in 1932 was appointed to a fellowship at Rensselaer Polytechnic Institute where he received the degrees of M.M.E. and D.M.E. Experimental work done at R.P.I. in heat transfer developed the methods used in the investigation reported in this paper. Dr. Pierson has been associated with Babcock & Wilcox Company since 1935.

[2] The Babcock & Wilcox Company's research program on convection heat transfer and flow resistance in cross flow of gases over tube banks includes (1) the experimental investigation of the influence of tube arrangement, (2) experimental investigation of effects of equipment size, and (3) correlation and utilization of new data regarding this subject. All three phases of the program are reported in this issue of the A.S.M.E. Transactions; the first by the author, the second by E. D. Huge, and the third by E. D. Grimison. See footnotes 3 and 4 on pp. 569 and 570, respectively, of this paper.

Contributed by the Heat Transfer Committee of the Process Industries Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers, to be held in New York, N. Y., December 6 to 10, 1937.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 W. 39th St. New York, N. Y., and will be accepted until December 10, 1937, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

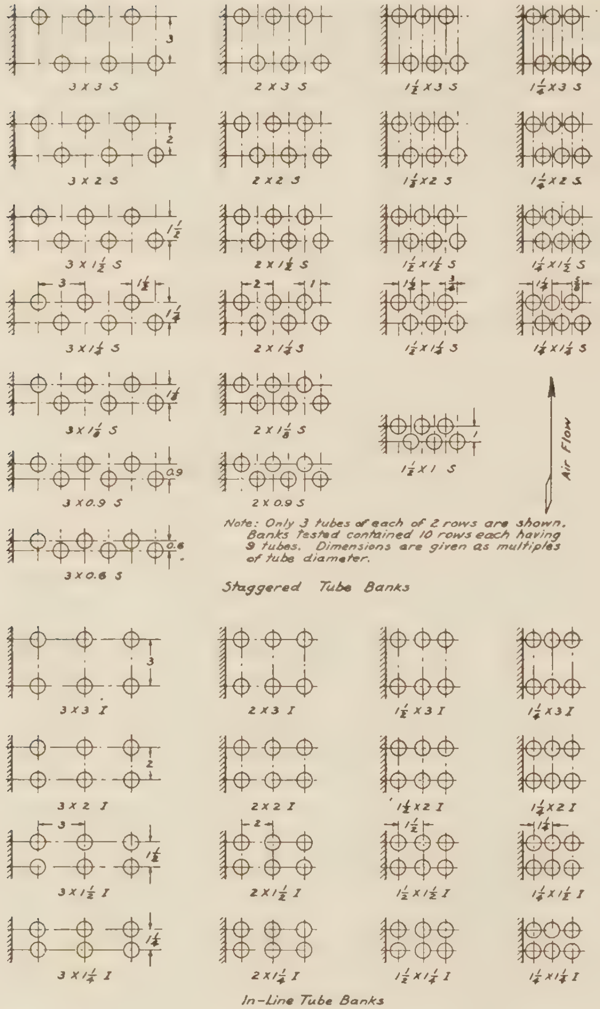

Fig. 1 Thirty-Eight Tube Arrangements for Which Flow-Resistance and Forced Convection Characteristics Were Determined for Reynolds' Numbers Between 2000 and 40,000

cities as characterize the prototype. This technique has been employed in flow studies in other fields with good results and earlier work by the author had developed it as applied to convection heat transfer.

The circulation of the air past a cooler makes it possible to evaluate the heat absorbed by the air, a useful heat-balance check on the heat input from the electrically heated tubes. To convert hourly heat per square foot of surface to air-boundary conductance, temperatures must be known with adequate accuracy; also to insure correct values of hourly heat, accuracy of other measurements must be assured within acceptable limits. The small scale of the apparatus imposed the need of special precautions.

It was early recognized that measurement of solid-surface temperatures was one of the chief sources of error. Detailed studies were made of several possible methods before deciding to determine tube temperatures from changes in electrical resistance of the tube itself. Tests made with thermocouples attached to the surface revealed a marked variation in temperature around the circumference of the tube, a condition verified by measurement of the very small bending distortion caused by heating the tube under test conditions. The electrical-resistance method gives a true average of the temperatures for the entire surface as has since been rigorously demonstrated by J. O. Jeffery in Bulletin 21 of the Cornell Experiment Station. Tubes were originally prepared for temperature measurement and calibrated by Leeds and Northrup Company who supplied the Kelvin bridge used in the observations. Details of these tubes and of the ordinary tubes are shown in Figs. 2 and 3. The calibrations were checked during the test program by independent methods discussed later where a sample calibration curve is shown in Fig. 19.

Comparable efforts were made to maintain absolute accuracy in other important observations. Details of these special tests of accuracy and unusual calibrations are given later but some require brief mention here. The thin-plate orifices were carefully calibrated and tests made to check air leakage in duct work between the tube bank and orifice. Air-stream temperatures were observed by five resistance pyrometers located under widely different conditions. Heat quantities were determined at four points, three of them completely independent. These heat quantities were all in good agreement as shown by the sample plot of them against air flow in Fig. 18. To be certain that the ten tube temperatures observed were representative of the entire bank, extended studies were made of heat distribution among the tubes and of air-flow distribution over them. Both distributions were repeatedly demonstrated to be uniformly good and unaffected by changes in test conditions. Similar flow explorations were also made before locating the impact tubes used in measuring the pressure drop across the bank, assuring observations free from error. All instruments used were, of course, calibrated and

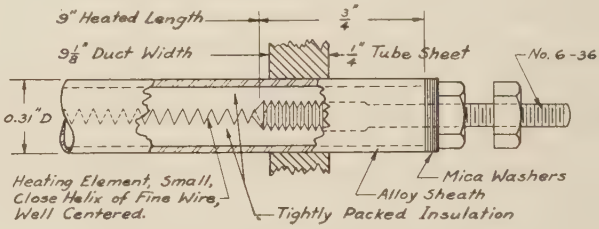

Fig. 2 Detail of Electrically Heated Tubes

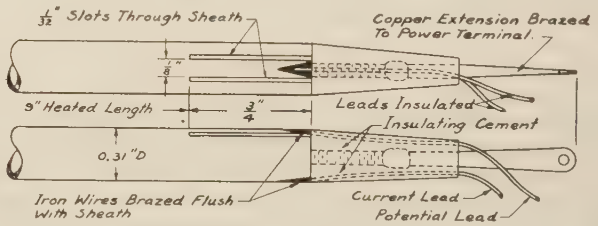

Fig. 3 Modification of Tube for Temperature Observations

checked periodically as explained in a section devoted to calibrations.

It is to be noted that the variation of heat input from 72,000 to 18,000 Btu per hr which changed tube temperatures, and the operation at zero heat input resulted in considerable new information on the effects of temperature, especially on flow resistance.

The method with its apparatus is well-adapted to much additional experimental investigation which it has not been possible to include in the present work. Thus, for example, air could be replaced by hydrogen or carbon dioxide as the circulated gas to secure valuable data on the effects of changes in physical properties of the fluid. Additional tests by the present methods on some irregularly arranged tube banks would be of interest as would be an extended study in which the flow through the bank varied from cross flow to longitudinal flow. Effects of bank depth,

flow approach conditions, and temperature levels have been partially studied in the present program but it is recognized that these studies are incomplete. With some modification of the apparatus temperatures could be varied over a wider range and information secured for transfer from hot air streams to cold tubes under conditions comparable to the present procedure.

The work here reported has covered in a comprehensive and systematic manner effects of tube arrangement and spacing. Because the extended nature of this investigation has resulted in a great mass of detailed information, this report, in the interest of brevity, is confined to only those results of major importance and general interest. For the same reasons it has been impract cable to detail the original observations from which the plots of results have been calculated although it is recognized that a complete tabulation would be of great value to those interested in further analysis of the data.

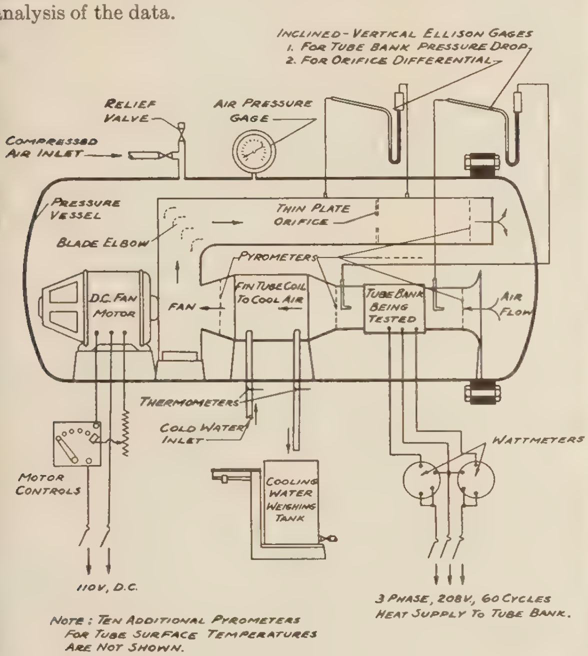

Fig. 4 Arrangement of Apparatus

Apparatus and Observations

The arrangement and operation of the apparatus are most readily followed by reference to Figs. 4, 5, and 6. All of the main working parts were enclosed in a specially designed pressure vessel 42 in. in diameter and 9 ft long. The specific weight of the air circulated through the tube banks could thus be controlled by varying the tank pressure between atmospheric and the design limit of 175 lb per sq in., unconditioned room air being supplied by a small reciprocating compressor. The tank was fitted with a flanged and bolted head to permit installation of the apparatus, but access was normally through the two standard manholes shown. One of these provided for care of the fan motor while the tube banks were set up in the apparatus through the other manhole after being largely assembled outside of the tank. Instrument, power, and water connections were carried through the tank shell in suitable airtight fittings.

The apparatus was essentially a small wind tunnel with added provisions for metering the total air flow and for cooling the air after it had been heated in the bank. Instruments were installed for the observation of all quantities as detailed in later paragraphs. Referring to Fig. 4, the high-specific-weight air approached the tube banks through the well-rounded tunnel inlet and the parallel-sided approach section. After absorbing heat in the tube bank it reached the Aerofin form of fin-tube cooling coil through the parallel-sided tunnel discharge having the same

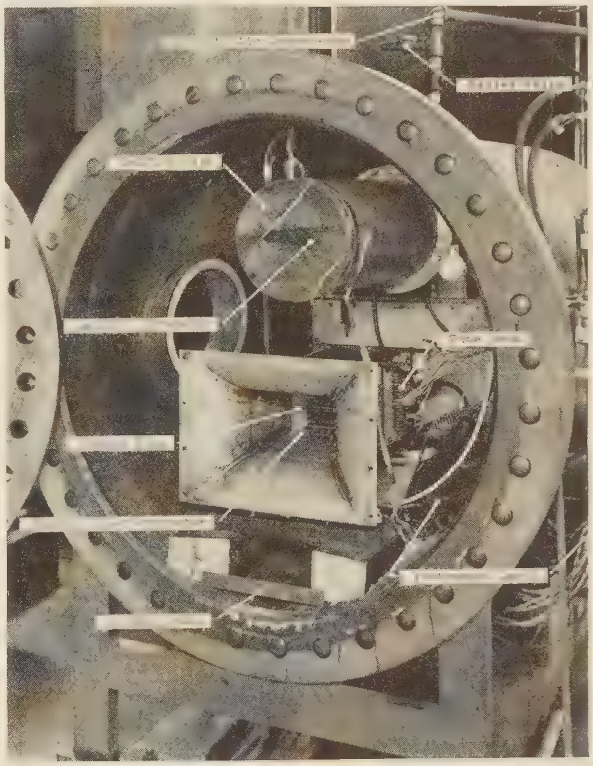

Fig. 5 Interior of Test Vessel

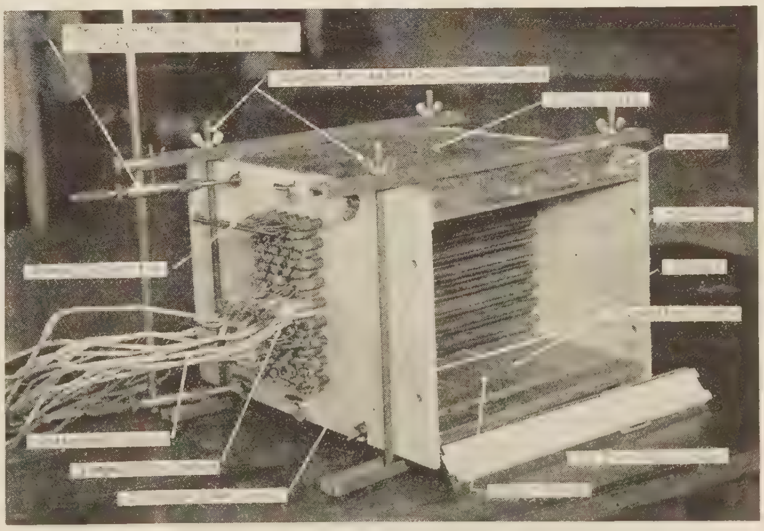

Fig. 6 Typical Tube Bank

flow area and shape as the approach section. The cooled air passed through the centrifugal fan (driven by a direct-connected variable-speed direct-current motor) and flowed to the thin-plate orifice through a blade elbow. The orifice pipe discharged the air into the main body of the pressure vessel from which it was recirculated. To accommodate the wide range of tube spacings tested, four sizes of tunnels were used in conjunction with three sizes of orifice plates for the 10-in. orifice pipe.

The tube banks were composed of 90 modified General Electric Calrod units 0.31 in. in diameter and having a heated length of $9^1/_8$ in. These units are composed of a small tight coil of resistance wire surrounded by a compacted insulation and a cylindrical sheath of alloy. Figs. 2 and 3 show details of these units. These sheaths were dressed to a smooth surface and then aged at a high temperature. The result was a smooth dark-colored surface which remained stable throughout the test program. The main series of tests was conducted with the heating elements of these tubes connected electrically in parallel groups of ten with three such parallel groups in series across each phase of the 208 v, 3-phase a-c supply. Such electrical connections supplied 72,000 Btu per hr to the banks while an alternate connection of five tubes in parallel with six parallel groups in series on each phase reduced the input to 18,000 Btu per hr. Experimental investigation proved this input to be evenly distributed between the tubes, the heating elements being well-standardized as to resistance and having low thermal coefficients of electrical resistance.

No external control of the heat transferred was installed. This condition led to considerable temperature variation within the banks at any given flow as well as with changes in air flow. Thus, tube temperatures during the program ranged from 250 to 750 F while air temperatures at the bank exit varied from 140 to 450 F. As these temperature variations were similar for all banks they are not inconsistent with the primary objective of determining the effects of tube arrangement. During later stages of the work, it became desirable to investigate the effects of temperature upon both convection heat transfer and flow resistance. Some tests were made using the alternate electrical connection described previously, but the available temperature range was too limited to be conclusive for heat transfer. In the case of flow resistance, the temperature range was extended by tests with air and tubes at room temperatures.

Efforts were made to provide as many checks of accuracy as possible in the apparatus itself. Thus, air temperatures were measured at five points: The tunnel inlet, tunnel exit, cooling-coil exit, orifice pipe, and main body of air, as shown in Fig. 4. The last three of these provide good checks upon the first, the resistance thermometers being installed under a variety of conditions. Heat supplied electrically to the bank was metered by two wattmeters in the three-phase supply line. This was checked against the heat absorbed by the water circulated through the cooling coils, the duct between the tube bank and the cooling coils being well-insulated as was the entire bank. Calculation of the same two heat quantities from the air-side data provided a good test of air-flow and temperature measurements. All of these quantities were in good agreement, a sample plot of them being shown in Fig. 18. Tube temperatures were observed in each of the ten rows, the observations always being consistently related. In addition to these tests of accuracy inherent to the apparatus, a great deal of time was devoted to independent studies. These are discussed in some detail in the latter part of this report along with the unusual calibration work.

Instruments were all grouped at a single table adjacent to the apparatus. A series of plug connections facilitated making the multiple temperature observations. Because of the small thermal capacity of the apparatus, conditions between test runs could be changed quickly. It was possible for a single observer to set up a tube bank and make nine to twelve test runs in about 16 hours.

Analysis

The observations have been converted into dimensionless groups commonly known as Nusselt numbers and friction factors, and these are shown as functions of Reynolds' number. These are defined by the three following dimensionless equations

$$\text{Nusselt number} = Nu = UD/k \quad \ldots\ldots [2]$$

$$\text{Friction factor} = f = \frac{10.84 \times 10^{-8} \rho \, \Delta P}{N\, G^2} \quad \ldots\ldots [3]$$

$$\text{Reynolds' number} = Re = GD/\mu \quad \ldots\ldots [4]$$

where U = gas-boundary convection conductance, Btu/[(hr) (sq ft) (deg F)]; D = tube diameter, ft; ΔP = gas pressure drop through bank, in. of water; G = gas mass flow through minimum free-flow area in bank, lb/[(hr) (sq ft)]; N = number of major restrictions in passing through bank; ρ = gas specific weight, lb per cu ft; k = thermal conductivity of gas, Btu/[(hr) (sq ft) (deg F/ft)]; and μ = absolute viscosity of gas, lb/[(hr) (ft)].

The major uncertainty in the application Equations [2], [3], and [4] to tube banks in cross flow is the question of the temperature at which to evaluate the physical properties of the gas as used in the equations.

For the present work the curves of Nusselt versus Reynolds numbers have been evaluated at the mean gas-boundary or film temperature, defined as the average tube temperature minus one half of the mean temperature difference. This follows the usual practice although there is evidence within the rather narrow temperature ranges of the present data that this procedure does not fully account for temperature effects in convection heat transfer.

In the case of the friction factors it was found necessary to select a new temperature for evaluation to correlate flow-resistance measurements made on a number of tube arrangements at three different temperature levels. Study of such sets of data for a large number of tube arrangements resulted in evaluation of the curves of the friction-factor versus Reynolds' number at a modified film temperature. This temperature is defined as tube temperature minus eight tenths of the mean temperature difference for staggered-tube arrangements, and as tube temperature minus nine tenths of the mean temperature difference for in-line arrangements. Evaluation at this temperature has greatly improved the correlation of flow-resistance measurements made at different rates of heat transfer in the same tube bank. A notable exception is found in certain in-line arrangements closely spaced in the direction of flow where the heating of the tubes apparently creates a flow condition differing markedly from that obtaining under isothermal conditions at lower Reynolds numbers.

The physical properties of air involved in the dimensionless groups have all been evaluated from data for dry air, investigation showing the effects of humidity to be negligible. Specific weight was based on the U. S. Bureau of Standards values for dry air. Sutherland's equation was used to evaluate viscosity while conductivity values were obtained from data given in the International Critical Tables, Vol. V.

No corrections for radiation have been applied to data for banks ten rows deep, the maximum calculated corrections being less than 1 per cent. Such correction has, however, been applied to results for the more shallow banks, amounting to some 10 per cent for a single row of tubes. All observations have been corrected for instrument calibrations.

Results

The results of the main series of tests in which tube arrange-

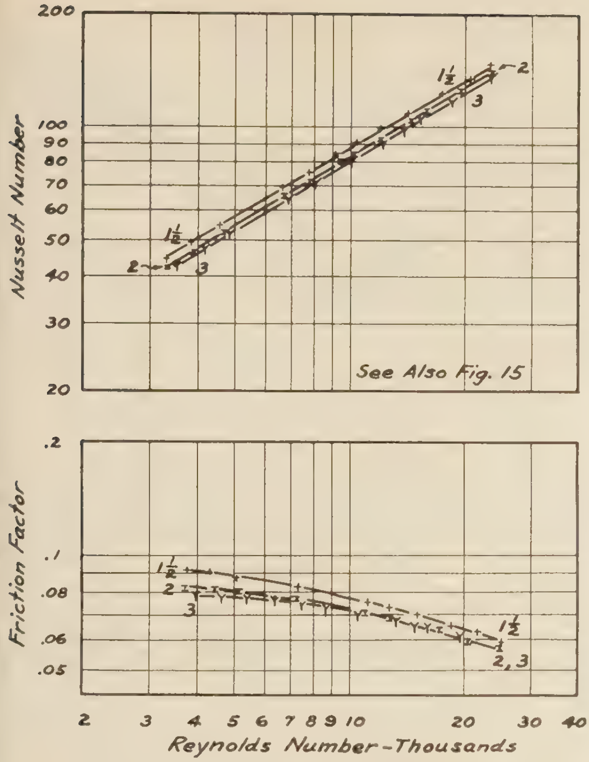

FIG. 7 RESULTS FOR 3-DIAMETER TRANSVERSE SPACINGS, STAGGERED ARRANGEMENT

(Figures on curves indicate tube pitch parallel to flow, expressed in tube diameters.)

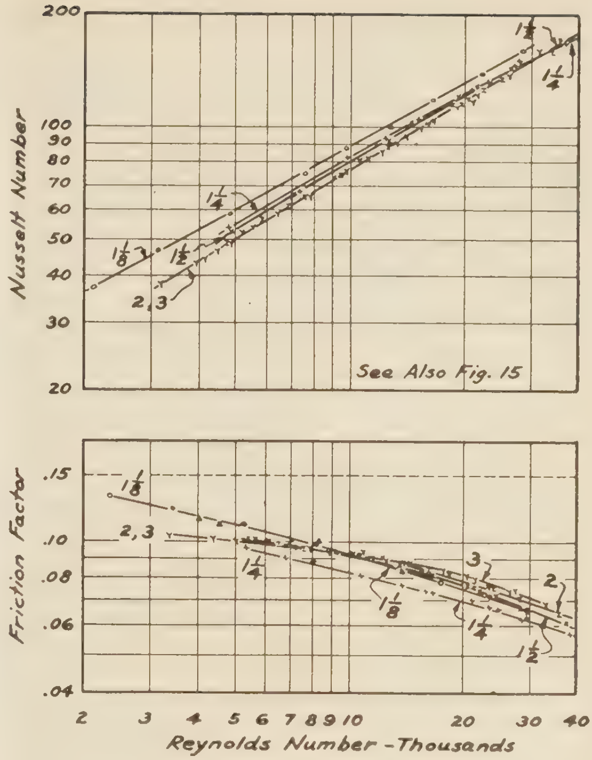

FIG. 8 RESULTS FOR 2-DIAMETER TRANSVERSE SPACINGS, STAGGERED ARRANGEMENT

(Figures on curves indicate tube pitch parallel to flow, expressed in tube diameters.)

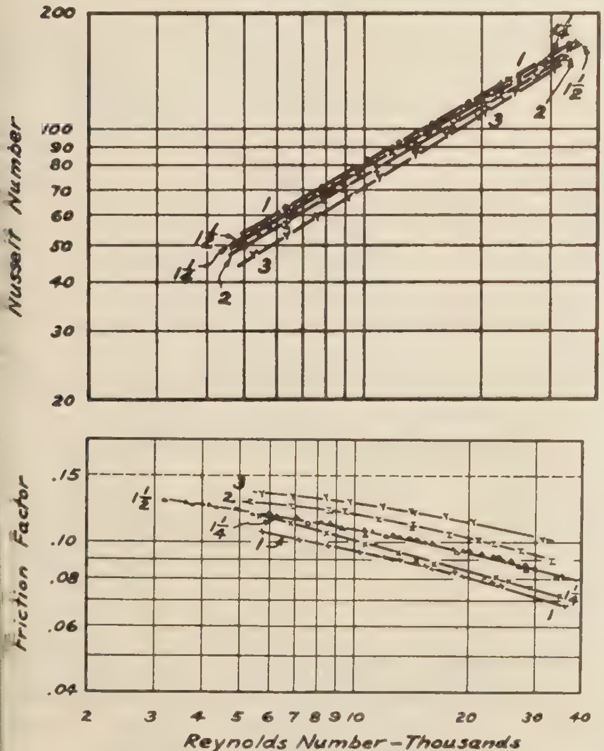

FIG. 9 RESULTS FOR $1\frac{1}{2}$-DIAMETER TRANSVERSE SPACINGS, STAGGERED ARRANGEMENT

(Figures on curves indicate tube pitch parallel to flow, expressed in tube diameters.)

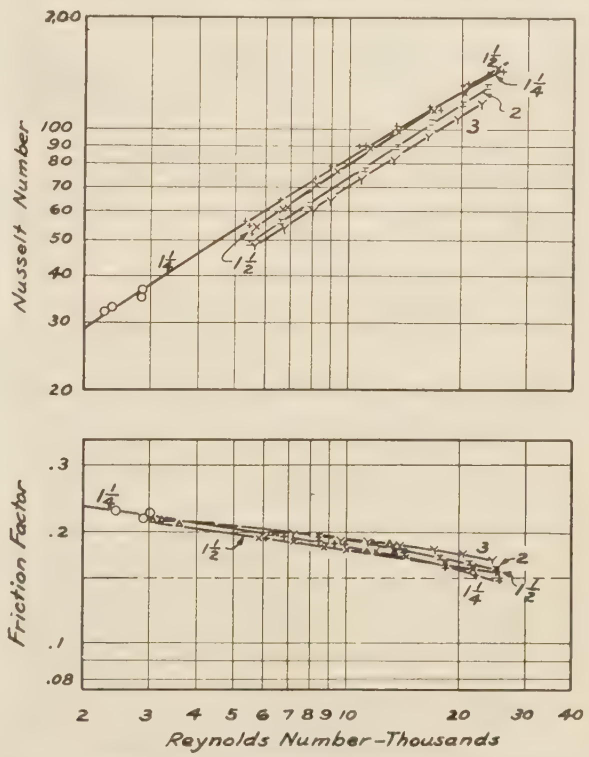

FIG. 10 RESULTS FOR $1\frac{1}{4}$-DIAMETER TRANSVERSE SPACINGS, STAGGERED ARRANGEMENT

(Figures on curves indicate tube pitch parallel to flow, expressed in tube diameters.)

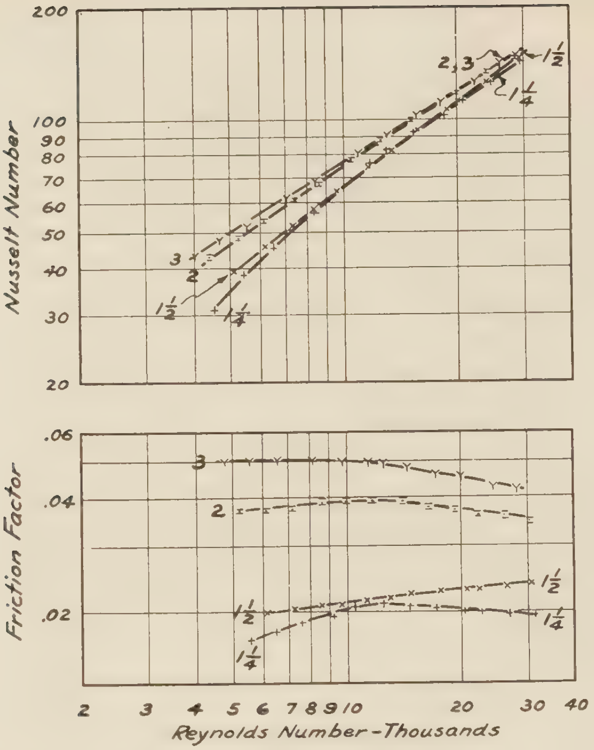

Fig. 11 Results for 3-Diameter Transverse Spacings, In-Line Arrangement

(Figures on curves indicate tube pitch parallel to flow, expressed in tube diameters.)

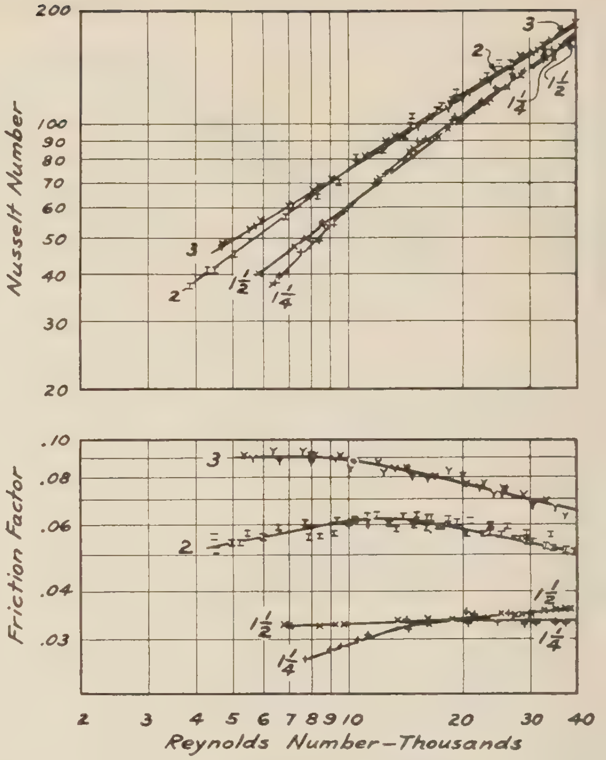

Fig. 12 Results for 2-Diameter Transverse Spacings, In-Line Arrangement

(Figures on curves indicate tube pitch parallel to flow, expressed in tube diameters.)

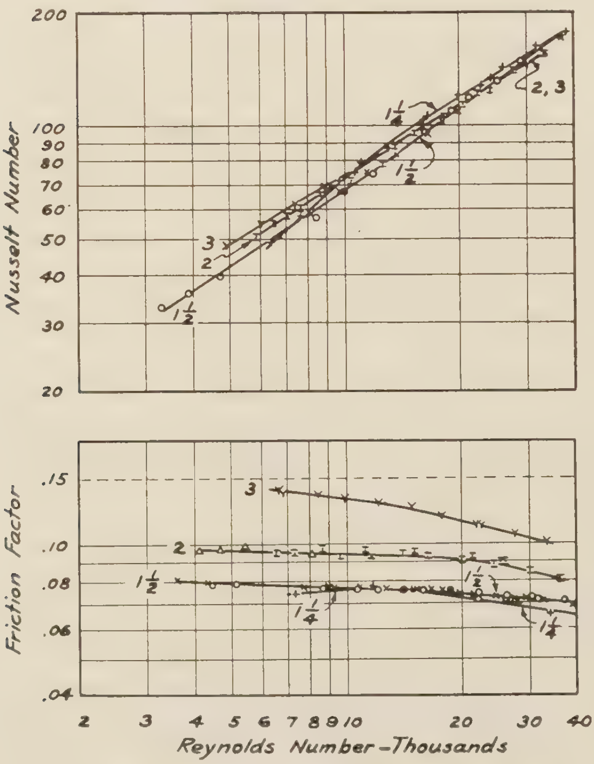

Fig. 13 Results for $1^1/_2$-Diameter Transverse Spacings, In-Line Arrangement

(Figures on curves indicate tube pitch parallel to flow, expressed in tube diameters.)

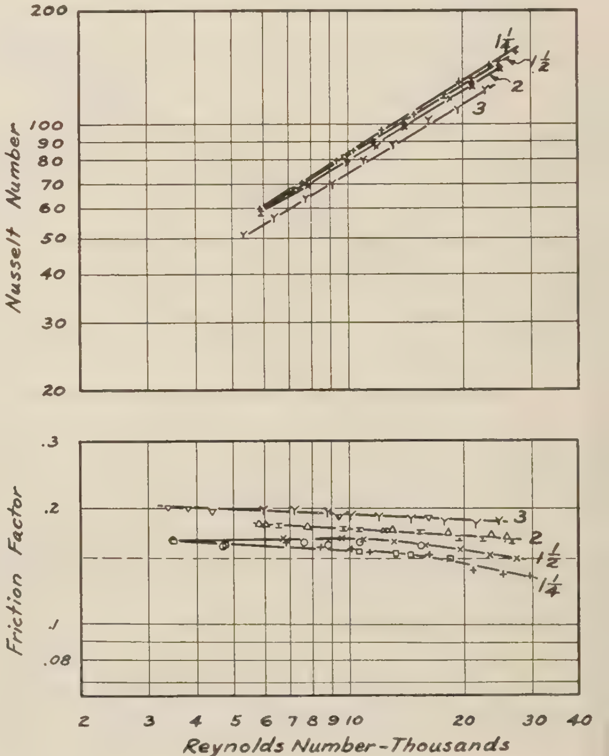

Fig. 14 Results for $1^1/_4$-Diameter Transverse Spacings, In-Line Arrangement

(Figures on curves indicate tube pitch parallel to flow, expressed in tube diameters.

ment was the chief variable are shown in the logarithmic plots of Figs. 7 to 15, inclusive. These plots show Nusselt numbers and

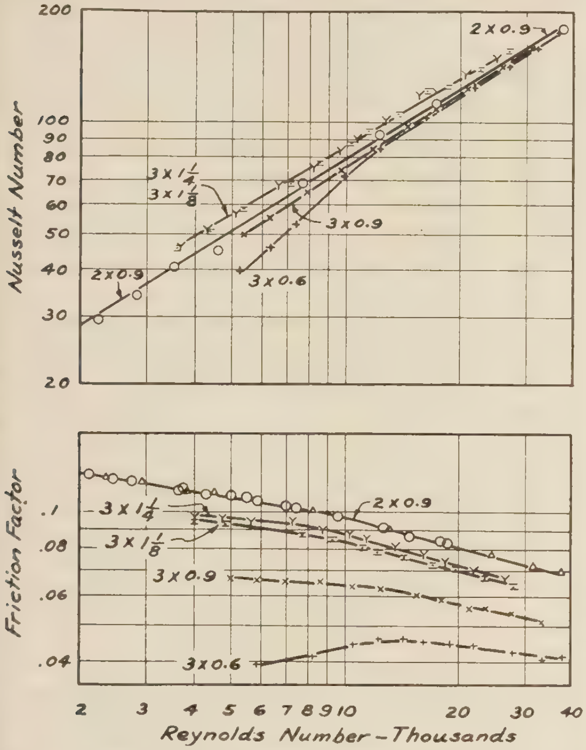

FIG. 15 RESULTS FOR STAGGERED BANKS IN WHICH MINIMUM FLOW AREA IS THROUGH DIAGONAL OPENINGS BETWEEN SUCCEEDING ROWS OF TUBES

(First figure on curve indicates tube pitch transverse to flow, expressed in tube diameters. Second figure is pitch parallel to flow.)

friction factors as functions of Reynolds' numbers. Curves have been drawn only within the actual range of test data and so represent accurately the range of conditions under which each tube arrangement was tested. Results have been grouped into the various families of transverse spacings tested, the spacings between centers being noted on the curve sheets in terms of tube diameters.

Study of these graphic results shows that the Nusselt numbers follow the generally accepted variation, increasing roughly as the two-thirds power of the Reynolds number, the actual exponent ranging from 0.55 to 0.80. It will be noted that tube arrangement affects the absolute value of the Nusselt number at a given Reynolds' number, as well as its variation with Reynolds' number. The variation with Reynolds' number may be influenced to some extent by the variation in tube temperatures associated with conducting the tests with a constant thermal current. For present purposes it is sufficient that for any given Reynolds' number all tube arrangements had substantially the same temperature conditions, and so seem to be directly comparable.

The friction factors in general remain constant or decrease as Reynolds' number is increased. Tube arrangement influences the variation with Reynolds' number as well as the absolute value of the friction factors. Changes in tube arrangement have a much more marked effect on the friction factors than on the Nusselt number. There is no apparent useful relationship between flow resistance and convection transfer rates, but their variations with Reynolds' numbers appear definitely related as reported in detail by Grimison.[2,3]

Study of the curves showing the relation of Nusselt numbers and friction factors to Reynolds' numbers indicates a change in flow conditions for some arrangements at a Reynolds number of about 10,000. For many tube arrangements, and particularly for in-line arrangements closely spaced in the direction of flow,

[3] "Correlation and Utilization of New Data on Flow Resistance and Heat Transfer for Cross Flow of Gases in Tube Banks," by E. D. Grimison, Trans. A.S.M.E., vol. 59, October, 1937, paper PRO-59-8, pp. 583–594.

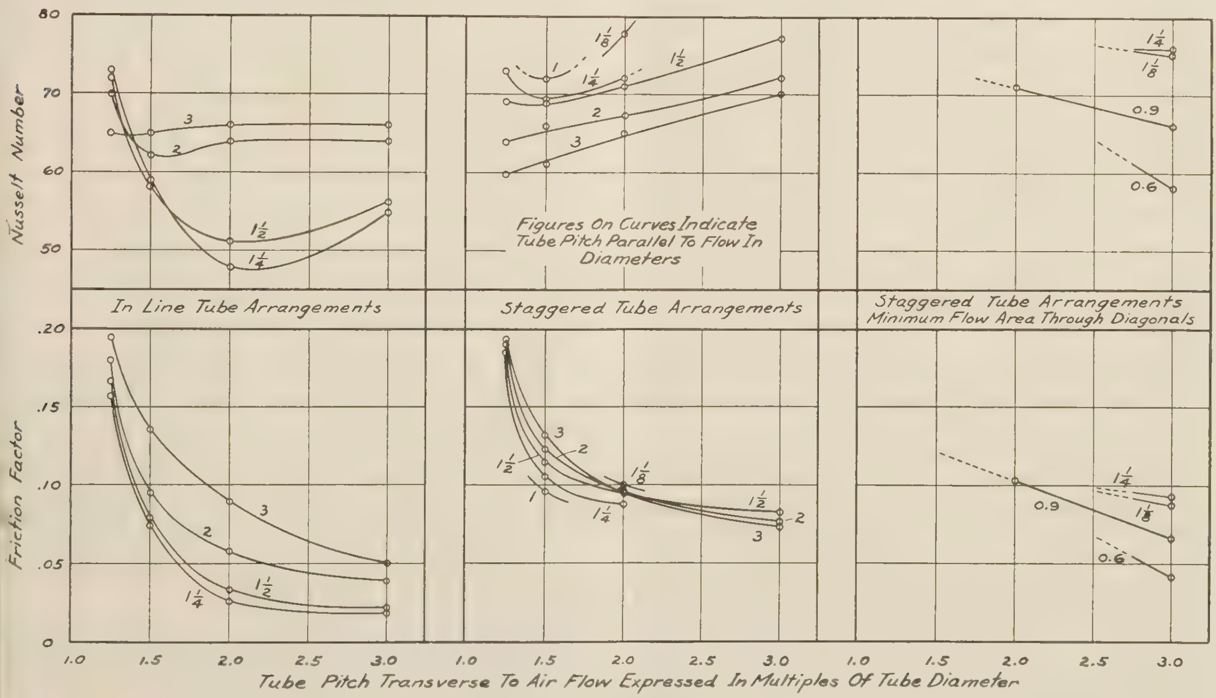

FIG. 16 EFFECTS OF TUBE ARRANGEMENT UPON NUSSELT NUMBER AND FRICTION FACTOR WHEN REYNOLDS' NUMBER IS 8000

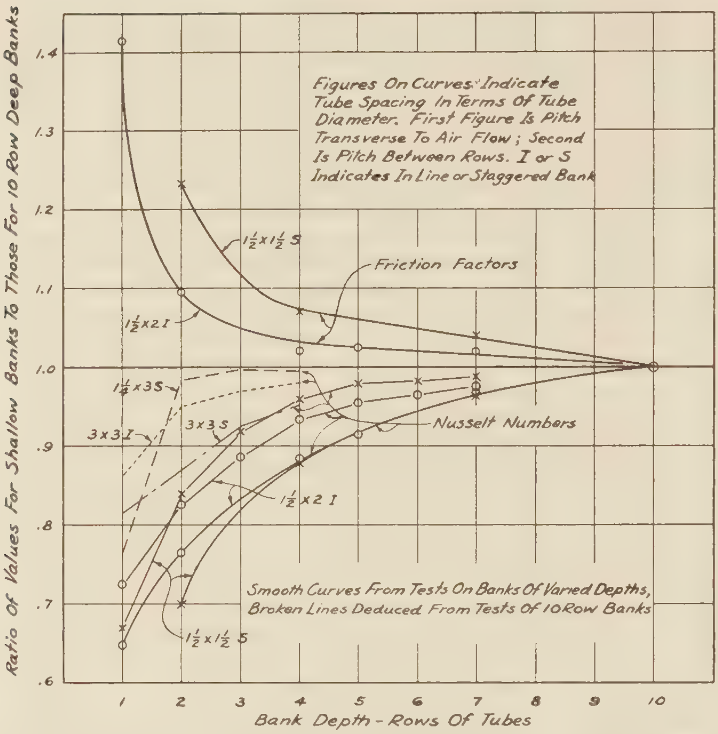

FIG. 17 EFFECTS OF BANK DEPTH ON NUSSELT NUMBER AND FRICTION FACTOR WHEN REYNOLDS' NUMBER IS 8000

there are changes in slope in both flow-resistance and convection curves at this point. It should be noted again that the Reynolds numbers used in plotting the two characteristics have not been evaluated at the same temperature, which may explain why the Nusselt and friction curves do not change slope at the same Reynolds number for a given tube arrangement. The effects of passing from one flow condition to another vary widely with tube arrangement, being least marked for the two extremes of spacings studied. While not fully investigated or analyzed, the condition must be borne in mind in applying the data, particularly if extrapolation beyond the test ranges is undertaken.

The systematic effects of tube arrangement upon both friction factors and Nusselt numbers is much more apparent when the foregoing data are plotted with reference to tube arrangement for constant values of Reynolds' number. Such a plot is shown in Fig. 16, spacings again being given in terms of tube diameters. There is no apparent useful relationship between flow resistance and transfer rates at a given Reynolds number although their variations with Reynolds' number appear related.

The variation in Nusselt number from row to row within several banks is shown in Fig. 17. These curves have been calculated from data for banks ten rows deep. Similar curves prepared from tests on two differently arranged banks of depths from one to ten rows are superimposed on Fig. 17. No satisfactory explanation has been found for the disagreement between these results and similar ones deduced from tests of the same arrangements in banks ten rows deep. Within this limit of accuracy, however, it is apparent that the variation of conductance within a tube bank depends upon the spacing of the tubes. The tests of two banks of varying depths also gave information concerning the effect of bank depth on friction factor. These results are also shown in Fig. 17, the friction factor increasing sharply for the more shallow banks. This is as might be expected as all friction factors used in this report are based upon the total observed pressure drop without correction for entrance and exit losses. The variation of friction factor with bank depth is probably different for each tube arrangement, but within the limits of the present data is not affected by changes in Reynolds' number. Likewise, the variation in gas boundary conductance within a given tube bank was found to be virtually independent of the Reynolds number within the range covered by the tests.

Flow resistances of many tube arrangements were measured for zero heat input, or isothermal conditions with cold air flowing over cold tubes. These observations were in addition to those made during tests with the normal heat transfer, but results are not reported here being in substantial agreement. For the same reason, flow-resistance and heat-transfer tests with the thermal current reduced to one fourth its normal value are also omitted.

ACCURACY

Throughout the execution of the test program, much effort was devoted to attaining and maintaining accuracy in all phases of the test work.

Huge,[2,4] reporting the results of an extensive independent investigation, has established the validity of the model theory. His tests were also planned as a basic measurement of accuracy in the work reported in this paper.

In addition a number of special tests and calibrations were made as a part of the present program, some of which merit discussion in this paper.

All of the usual precautions were observed in the installation of the service orifice, but because of the necessarily crowded conditions it was decided to investigate the accuracy of flow-rate observations. For this purpose an orifice flow meter consisting of a 16-ft approach and a 4-ft discharge section of 10-in. pipe was constructed. This was connected in series with the service orifice and simultaneous observations of flow made over a wide range of rates, using different orifice sizes. The results were consistent in indicating a correction of about 1 per cent, varying somewhat with the size of the service orifice.

A further investigation of flow measurement was made by attaching the orifice flow meter to the tunnel inlet section, comparing the measured flow with the corrected value indicated by the service orifice. This procedure revealed a small amount of leakage in the duct work which could not be entirely eliminated. This leakage was evaluated in terms of pressures following the bank and preceding the orifice, and all data were corrected to a true flow rate through the bank. The maximum correction, including that for poor location of the service orifice, was about 3 per cent with an average correction well below that figure.

Independent verification of these leakage corrections was secured by providing an air-tight circuit between the service-orifice discharge and the tunnel inlet. Thus, all of the air metered entered the tube bank, eliminating all flow corrections except those for service-orifice calibration. Tests were conducted under these conditions over the entire test range and found to be in good agreement with tests of the same tube arrangements made under normal testing conditions. Additional evidence of accuracy is found in the excellent agreement of duplicate tests made with 3-in. and 6-in. orifices and in the good heat balances.

[4] "Experimental Investigation of Effects of Equipment Size on Convection Heat Transfer and Flow Resistance in Cross Flow of Gases over Banks of Tubes," by E. C. Huge, Trans. A.S.M.E., vol. 59, October, 1937, paper PRO-59-7, pp. 573–581.

Tests were repeated on certain arrangements with several other tests intervening between duplicate runs. At one time difficulty in reproducing results on some close spacings as well as poor agreement with Huge's tests[4] led to an investigation of the effects of variations in free-flow area through the banks. It was found that for the most closely spaced tube arrangements variations as small as $^1/_{64}$ in. in the duct width at the bank had a perceptible effect on test results. Provision was made to control these dimensions to 0.01 in. by a system of screw arrangements, spacers and micrometer measurements, and no further trouble from this source was encountered. In this connection it must be noted that no half tubes were used to fill out the tube rows of staggered arrangements, which may account for the extreme sensitivity to variations in flow area. Studies of flow distribution proved it to be good in all ranges of spacings tested, provided the flow areas were properly adjusted. Such flow explorations were made by several methods, including pitot tubes, thermocouples, temperature measurements in the bank, and observation of draft loss across several tunnel sections.

Some indication of over-all accuracy in observations is contained in heat balances when based on completely independent measurements as is possible here. The primary observation of heat transferred was made by metering the electric input to the tubes. This was compared with the absorption indicated by observation of air-stream temperature rise and total air flow. A secondary calculation of air-stream heat quantities was based upon the temperature drop in passing through the fin-tube cooling coils The third independent evaluation of heat transferred was based on the temperature rise of the cooling water and its rate of flow. All four of these quantities were in general good agreement as illustrated in Fig. 18. Thus, it is proved that all of the heat supplied as electrical energy was absorbed by the air stream. Moreover, since the possibility of compensating errors is extremely remote, these heat balances present additional evidence of accuracy in determining air flow and temperatures.

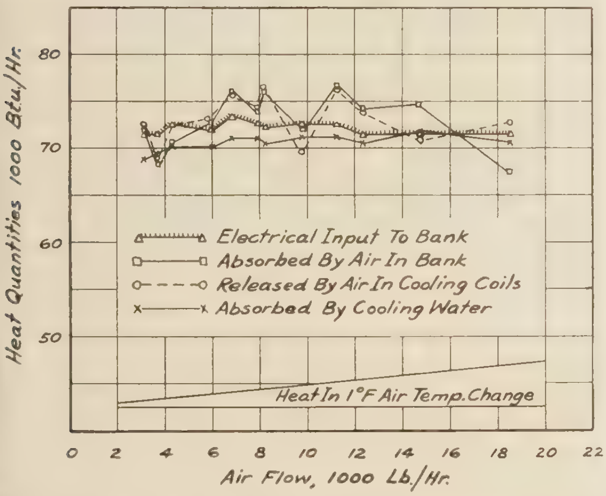

FIG. 18 HEAT QUANTITIES AS EVALUATED INDEPENDENTLY FROM ELECTRICAL INPUT, AIR FLOW AND TEMPERATURES, AND WATER FLOW AND TEMPERATURES. FROM TESTS OF 3 × 3S ARRANGEMENT

The humidity of samples of air bled from the pressure vessel was determined during one test series. It was found to remain substantially constant over the entire range of pressures, the relative humidity being well under 50 per cent at the bank inlet conditions. Any effect of humidity was within the accuracy to which the physical properties of air are known and all calculations have accordingly been based on the properties of dry air.

As a check on tube-temperature measurement, tests were conducted on a single tube with air in cross flow. These were in good agreement with the large body of information available for the simple case of convection heat transfer. Additional evidence of accuracy of tube-temperature measurements is discussed in later paragraphs.

Unusual Calibrations

Calibrations were made of all important instruments used in the test work. These calibrations were rechecked during the program to maintain accuracy. In the case of such instruments as wattmeters, thermometers, thermocouples, and pressure gages the calibration procedures are well standardized and so are not discussed in this paper.

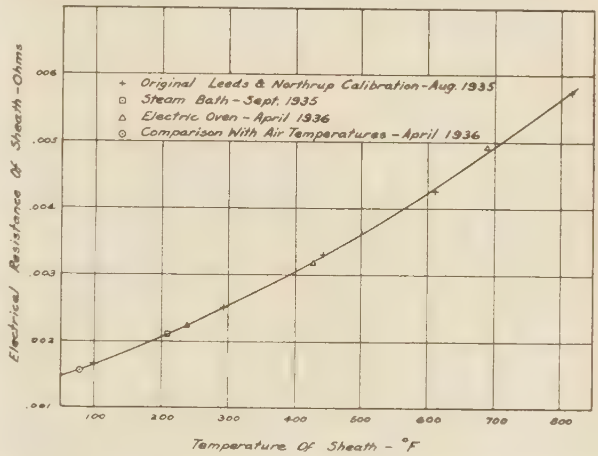

FIG. 19 SAMPLE CALIBRATION CURVE FOR TUBE TEMPERATURES

Ten of the tubes used in the banks were prepared and calibrated for observation of tube-surface temperature by Leeds and Northrup. A Kelvin bridge was arranged to read the electrical resistance of that part of the tube sheath actually heated and in the air stream. Details of these electrical connections are shown in Fig. 3. During the test program it became necessary to check these for possible changes in calibration and to prepare replacements. For this purpose a small electric furnace was constructed by experimentally distributing heating coils to produce a uniform temperature over the central 10 in. of a 24-in. fused silica tube, the inside diameter of which was $^3/_4$ in. The furnace was well insulated and provided with controls to vary the temperatures up to 1000 F. Tubes were placed in the furnace one at a time and heated to equilibrium at a number of temperature levels. Temperatures were read from four carefully calibrated thermocouples distributed along the tube while the sheath resistance was read from the same Kelvin bridge used for all test observations. Readings were taken in regular rotation until constant conditions were indicated for 20-min periods at each temperature level. These check calibrations revealed no change from the original during the period covered by the tests. Additional independent checks of the calibrations were made by measuring sheath resistance while immersed in a wet-steam bath at atmospheric pressure. The temperature tubes were also checked periodically by comparison with indications of the air stream pyrometers when pressure-drop tests were being run with cold air. A sample calibration curve showing the typical good agreement between these methods is shown in Fig. 19.

The air-stream resistance grid pyrometers were calibrated by immersion with a standardized mercury thermometer in a constantly agitated, slowly heated oil bath. Alternate readings of

resistance and temperature were plotted against time and checked against equilibrium data. Two pyrometers were calibrated over the entire temperature range and checked after two months of severe service. No change was found, so additional pyrometers made from the same spool of nickel wire were calibrated over a limited temperature range and extended by comparison with the original calibrations. The reliability of the calibrations is indicated by good agreement of temperatures indicated under isothermal conditions and by the satisfactory heat balances. The water-side heat absorption proved all the heat to be entering the air stream, so calculated air temperatures leaving the bank were employed in the analysis.

Observations Made and Ranges of Variables

1 Air-stream observations:

(*a*) *Temperatures.* Thermometers were of the resistance-grid type made of No. 29 nickel wire, and were read by Leeds and Northrup Kelvin-bridge ohmmeters. The range of the temperatures entering the bank was 80 to 110 F, while leaving the bank the range was from 140 to 450 F.

(*b*) *Total Air Flow.* Total air flow was measured by 3-in. to 6-in. thin-plate orifices in a 10-in. pipe with static taps located one pipe diameter upstream and one-half diameter downstream. Differentials were read from an Ellison inclined-vertical draft gage having the first 3 in. of differential on a 15-in. inclined scale. The range of flow was from 300 to 21,000 lb of air per hr.

(*c*) *Bank Flow Resistance.* Bank flow resistance was measured by impact tubes centered in tunnels upstream and downstream of the tube bank. The differential was read from an Ellison gage similar to the orifice gage for measuring total air flow. The range of this resistance was from 0.1 to 12 in. of water.

(*d*) *Air Pressure.* The air pressure was read from a calibrated Bourdon-type gage with a range from 0 to 100 lb per sq in. gage.

2 Tube-bank observations:

(*a*) *Bank Input.* The heat input to the tube bank was read from two Weston switchboard wattmeters connected to the 3-phase alternating-current supply, and was checked against air-stream and water heat absorptions. The input was normally 72,000 Btu per hr, although some tests were made at 18,000 Btu per hr.

(*b*) *Tube Temperatures.* Alloy sheaths of ten tubes were calibrated as resistance thermometers, leads being arranged to measure only the temperature of the section in the air stream. The temperatures were read from a Kelvin-bridge ohmmeter. The normal range of the tube temperatures was 250 F to 750 F; with reduced input, 150 to 600 F.

(*c*) *Tube-Bank Dimensions.* The tube diameters were read at random by a micrometer, the final value used being an average of over 100 measurements. Other dimensions were read by calipers and a steel scale graduated in hundredths of an inch. Diameter of the tubes was 0.31 in.; other dimensions were 3 to 9 in.

3 Water heat-absorption measurements:

(*a*) *Temperatures.* Calibrated mercury thermometers were located close to the pressure-vessel walls in the water inlet and outlet lines. The range of the temperatures in the inlet line was 40 to 70 F, while the range of those in the outlet line was 90 to 130 F.

(*b*) *Water Flow.* Water was collected in a weighing tank for 20-min periods during each run and weighed on platform scales. The rate of the flow ranged from 1000 to 2000 lb per hr.

Additional general observations such as barometer height and room temperature were also recorded during each test series. All important instruments were calibrated against laboratory standards or fixed points, details of unusual calibrations being given previously in this report. Readings were taken always in a fixed rotation and at regular time intervals until three consecutive consistent sets were obtained for a given set of conditions. The normal test procedure was to attain thermal equilibrium at the maximum air pressures used before taking any readings. Conditions between test runs were changed by varying the fan speed, reducing the air pressure, or both; ample time was allowed to restore equilibrium before recording more data. The apparatus was very stable in operation at all rates and a minimum of difficulty was experienced in maintaining steady conditions while recording observations.

Experimental Investigation of Effects of Equipment Size on Convection Heat Transfer and Flow Resistance in Cross Flow of Gases Over Tube Banks

By E. C. HUGE,[1] BARBERTON, OHIO

To establish the effect of tube size as a factor in heat transfer and pressure drop for gases flowing transversely over tube banks[2] of a given tube arrangement, new experimental determinations have been made for several tube sizes, and results compared with those of Pierson[3] who used very small tubes of 0.31 in. diameter. The arrangements used included some of those studied by Pierson[3] and these tests on a larger scale have served as a check on the accuracy of his small-scale results with a different method and different imposed conditions, and the applicability of model results to full-scale equipment, the two series forming parts of a research program[2] of the Babcock & Wilcox Company.

Nine tube arrangements as shown in Fig. 1, defined by center spacings in the direction of flow and transversely, were used with tubes of both $1/2$ in. diameter and $11/16$ in. diameter, the spacing ranging from a minimum of $1\,1/4 \times 1\,1/4$ diameters, to a maximum of 2×3 diameters with ten rows in the direction of flow both in-line and staggered, and ten tubes wide except for one case of eight tubes wide. In addition, three arrangements of tubes of 2-in. diameter were used in banks of ten rows deep and from nine to fifteen tubes wide.

For the banks of smaller tubes, $1/2$ in. and $11/16$ in. diameter, heat was transferred from hot gases outside to cold water inside, while for the banks of 2 in. tubes, heat was transferred from condensing steam inside to air outside.

Values of gas-boundary conductance and pressure drop were found for several identical tube arrangements and spacing and three tube sizes of $1/2$ in., $11/16$ in., and 2 in., with heat flow inward and outward; these are found to be consistent with those for the Pierson[3] model tubes of 0.31 in. diameter.

Heat-transfer rates are reported in terms of the Nusselt number, and pressure drop in terms of the Fanning equation for friction factor, both of them in relation to the Reynolds number. For values of the Reynolds number in the range of commercial practice, the Nusselt number varies as the 0.61 power of the Reynolds number, and the friction factor is not so regularly related to the Reynolds number but varies with the tube arrangement. Values of both gas-boundary conductance and friction factor for a given tube arrangement are consistent for all tube sizes from the full-scale 2-in. tubes down to the Pierson[3] model tubes of 0.31 in., including the intermediate sizes of $1/2$-in. and $11/16$-in. tubes, thus confirming the validity of the principle of similarity applied to tube banks in spite of some departure from true geometric similarity in the ratio of length to diameter or to intertube space for the range of Reynolds' numbers tested, 2000 to 70,000.

Test Method

FOR COMMERCIAL work of design and of calculating expected performance of a given design of tube-bank apparatus, such as used in steam boilers, it is necessary to have a means of calculation that will account for all variables including size of tube and tube spacing. Direct experimental measurement of heat-transfer rate between fluids inside and outside of the tubes for the ranges of sizes and spacing in use being too expensive an undertaking, an alternate method becomes necessary. The effect of tube spacing or arrangement in banks may be directly determined in model apparatus at relatively low cost, and this has been done by Pierson[3] for thirty-eight arrangements, but before such model results may be applied to full-scale apparatus and flow conditions it is necessary to establish the effect of size.

Passing from the model tubes electrically heated and with mean tube-face temperature directly measured for fixing the conductance of the gas boundary to full-scale tubes, a change in experimental procedure becomes necessary. As tube size increases, the cost of apparatus increases rapidly and fewer arrangements may be tested within reasonable cost limits. At the same time, a different mode of heating must be used with corresponding difference in the determination of tube-face temperature needed

[1] Engineer, Research and Development Laboratory, Babcock & Wilcox Company. Jun. A.S.M.E. Mr. Huge was graduated in 1930 from the Case School of Applied Science with a B.S. degree in mechanical engineering and since has been with the Babcock & Wilcox Company.

[2] The Babcock & Wilcox Company's research program on convection heat transfer and flow resistance in cross flow of gases over tube banks includes (1) the experimental investigation of the influence of tube arrangement, (2) experimental investigation of effects of equipment size, and (3) correlation and utilization of new data regarding this subject. All three phases of the program are reported in this issue of the A.S.M.E. Transactions, the first by O. L. Pierson, the second by the author, and the third by E. D. Grimison. See footnote 3 on this page and footnote 4 on p. 580 of this paper.

[3] "Experimental Investigation of the Influence of Tube Arrangement on Convection-Heat Transfer and Flow Resistance in Cross Flow of Gases Over Tube Banks," by O. L. Pierson, Trans. A.S.M.E., vol. 59, October, 1937, paper PRO-59-6, pp. 563–572.

Contributed by the Heat Transfer Committee of the Process Industries Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers, to be held in New York, N. Y., December 6 to 10, 1937.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1937, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

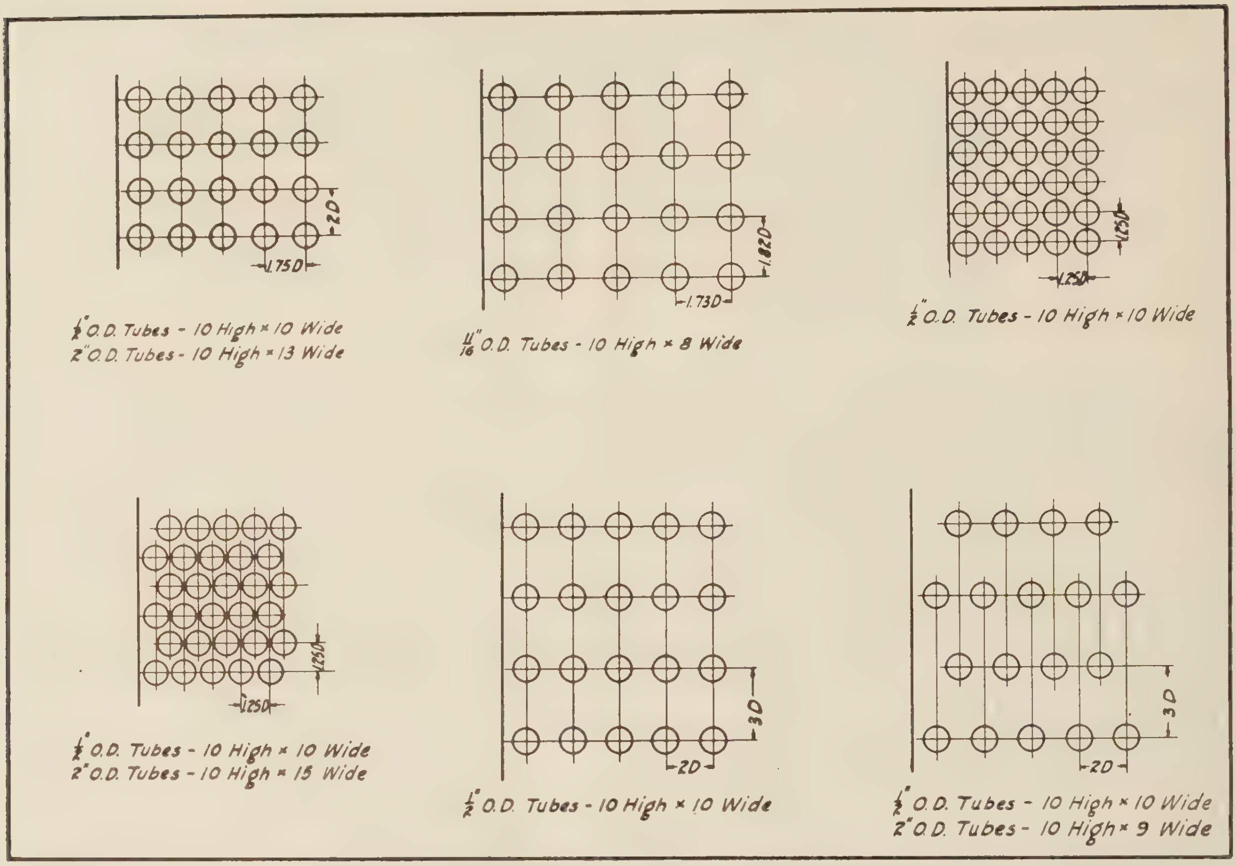

Fig. 1 Typical Sections of Tube Banks for Experiments

to convert hourly heat per square foot to gas-boundary conductance, and to measure the hourly heat itself.

The three tube sizes used were tested in the combinations of arrangement shown in Fig. 1 and tabulated in Table 1 in two apparatus designs, one termed "small scale" for the $^1/_2$-in. and $^{11}/_{16}$-in. tubes, and the other termed "full scale" for the 2-in. tubes. In the spacing designation in Table 1 the first number is the center-to-center distance between tubes in tube diameters, measured across the gas stream, that is, transversely. The second number is the spacing in the direction of flow. The term "wide" in the third column of Table 1 refers to the number of tubes transversely and "high" refers to the number in the direction of gas flow. Staggered arrangement differs from in-line arrangement by displacing alternate rows of tubes, one half of a tube space. The tube spacing includes one $1^1/_4 \times 1^1/_4$ which is considered a minimum, and one 2×3 which is considered a maximum, with some intermediate ones.

TABLE 1 TUBE SIZES AND BANK ARRANGEMENTS

Outside tube diameter, in.	Spacing, tube diameters	Arrangements	Width and height of banks, tubes
Small-scale apparatus:			
$^1/_2$	$1^3/_4 \times 2$	In-line	10×10
$^{11}/_{16}$	$1^3/_4 \times 1.82$	In-line	8×10
$^1/_2$	$1^1/_4 \times 1^1/_4$	In-line	10×10
$^1/_2$	$1^1/_4 \times 1^1/_4$	Staggered	10×10
$^1/_2$	2×3	In-line	10×10
$^1/_2$	2×3	Staggered	10×10
Full-scale apparatus:			
2	$1^3/_4 \times 2$	In-line	11×10
2	$1^1/_4 \times 1^1/_4$	Staggered	15×10
2	2×3	Staggered	9×10

Two different methods of establishing heat transfer, of measuring hourly heat, and of determining tube-face temperature were used, one for the two small tube sizes of $^1/_2$ and $^{11}/_{16}$ in., the other for the large 2-in. tubes.

The banks of small tubes in the small-scale apparatus were swept transversely by a mixture of air and products of combustion in adjusted ratios to predetermine the gas temperature at about 600 F, the mixture containing from about 1 to 2 per cent water vapor and 1 per cent CO_2 (Orsat), and water flowed inside of the tubes in general countercurrent relation. The hourly heat transferred was calculated from water quantities and temperature rise, and also from gas quantities and temperature drop corrected for casing and other losses, so a heat balance was available. Tube temperatures were directly determined for inlet and outlet tubes by thermocouples, but in addition tube temperatures were calculated from the over-all resistance, the reciprocal of the over-all conductance, by subtracting the resistance of the water boundary and that of the tube metal.

The water-side conductance was determined by use of a modified McAdams formula

$$U_w = 0.0270 \frac{k^{0.6} c_p{}^{0.4}}{\mu^{0.4}} \frac{G^{0.8}}{D^{0.2}} \quad \ldots\ldots\ldots\ldots [1]$$

where U_w = water-side conductance, Btu/[(hr)(ft²)(deg F)]; k = thermal conductivity of water, Btu/[(hr)(ft²)(deg F/ft)]; G = water mass flow, lb/[(hr)(ft²)]; μ = absolute viscosity of water, lb/[(hr)(ft)]; D = inside diameter of tube, ft; c_p = specific heat of water, Btu/[(lb)(deg F)].

The thermal conductivity k for steel at operating temperatures

was taken as 25.2 Btu per sq ft hr and per deg F per ft of thickness. The thermal resistance of the tube wall was calculated by use of the logarithmic mean tube-wall relation

$$R_s = \frac{1}{U_s} = \frac{D_o \log_e \dfrac{D_o}{D_i}}{2k} \qquad [2]$$

where R_s and U_s = resistance and conductance of steel referred to outside surface; D_o = outside diameter, ft; D_i = inside diameter, ft; and k = 25.2 Btu/[(hr) (sq ft)(deg F/ft)].

The banks of full-size 2-in. tubes were swept by cool air transversely which was heated by dry steam condensing inside, in most cases at 18 lb per sq in. abs, so hourly heat could be calculated from the air quantity and temperature rise, and also from the condensate quantity and latent heat above condensate temperature; therefore, a heat-balance check could be made on the accuracy of results. It is well established that under such conditions the tube temperature is that of saturated steam for the pressure used, but this was verified experimentally. Incidentally, the fact that the controlling thermal resistance is on the air side, insures equality of tube temperature around the circumference at the steam saturation value.

The conductance of the gas boundary is reported in terms of the Nusselt number which is proportional to the nth power of the Reynolds number

$$\frac{UD}{k} = C\left(\frac{\rho VD}{\mu}\right)^n \qquad [3]$$

where U = gas-boundary convection conductance, Btu/[(hr) (sq ft) (deg F)]; D = tube diameter, ft; k = thermal conductivity of gas, Btu/[(hr) (sq ft) (deg F/ft)]; C, n = constants; ρ = specific weight of gas, lb per cu ft; V = velocity of gas, ft per hr; μ = absolute viscosity of gas, lb/[(hr)(ft)]; UD/k = Nusselt number, dimensionless; and $\rho VD/\mu$ = Reynolds' number, dimensionless.

According to the principle of similarity, when flow conditions are geometrically similar as to arrangement, proportions, and tube surface, the Nusselt number will be the same for a given value of the Reynolds number.

Pressure drops in the air stream flowing across the tube bank are reported in terms of the dimensionless ratio of the form of the Fanning equation, which is proportional to the *Xth* power of the Reynolds number.

$$f = \frac{10.84 \times 10^{-8}\ \Delta P \rho}{NG^2} = C'\left[\frac{GD}{\mu}\right]^X \qquad [4]$$

where f = friction factor, dimensionless; C', X = constants; ΔP = pressure drop, in. of water; N = number of major restrictions in passing through bank; and other notation is same as given for Equation [3].

This is the generally accepted method of comparing resistance to flow, across tube banks for example, since, with similar conditions of arrangements and proportions and for a given value of the Reynolds number, the expression for f which includes pressure drop will have the same value.

In applying the observations to Equation [3], the relation of the Nusselt to the Reynolds number, the gas temperature at which to evaluate the variables was taken as the mean gas boundary or film temperature which is the tube temperature plus or minus one half of the log mean temperature difference (plus for gas cooling and minus for gas heating).

The Fanning relation, Equation [4], with its corresponding Reynolds' number has been calculated on a revised film-temperature basis which is tube temperature plus or minus 0.8 or 0.9 of the log mean temperature difference (0.8 for staggered and 0.9 for in-line bank). The temperature basis for the Fanning relation is empirical and its use is justified by its superior correlation of data.

The physical constants of viscosity μ_f and thermal conductivity k_f were evaluated on a mean-film-temperature basis. The physical constant μ_{fs} and density ρ_{fs} were evaluated on a revised mean-film-temperature basis as previously outlined.

The values for absolute viscosity in lb per ft hr were taken from a curve of Sutherland's equation which is checked by experimental points from Bremond.

The values of thermal conductivity in Btu per sq ft hr deg F per ft of thickness were taken from a curve k = constant $\times \mu \times c_V$ which is checked between 100 and 400 F by experimental points from the International Critical Tables, Vol. 5.

The density ρ of air was calculated from basic values of density given by the U. S. Bureau of Standards and International Critical Tables.

The gas-boundary conductance results and those for friction factor of the Fanning equation, reported graphically on a base of Reynolds' number, are discussed later.

Apparatus

Small-Scale Apparatus. The small-scale apparatus for testing the banks of 1/2-in. and 11/16-in. tubes is shown in Fig. 2. Room-temperature air was supplied by means of a Sturtevant pressure blower which discharged to a 21-in. diameter pipe within which a Bailey thin-plate orifice was located according to recommendations of the Bailey Meter Company. The guaranteed accuracy of the thin-plate orifices when used with an accurate manometer is plus or minus 2 per cent; however, it is believed that considerably less deviation was the rule as checks made by overlapping orifice sizes between reasonable values of orifice differential gave no indication of any measurable deviations.

The air stream was heated by direct contact and mixture with products of combustion of a gas-fired furnace as also shown in Fig. 2. The air stream was separated after it had been metered and a small portion of it passed to the gas furnace for the purpose of supporting combustion, the remainder being directed to a gas-mixing chamber located at furnace exit.

The products of combustion of the gas-fired furnace and cold air were mixed within the mixing chamber in the amount required to give the desired gas-inlet temperature below the tube bank being tested.

For the water circuit of the apparatus water was taken from a Scaife "base-exchange" water-treating system to a water heater and from the heater the flow was divided into two circuits so that part of the water passed through a radiation screen circuit and part through the tube arrangement being tested. The water inlet of the element being tested was on the top of the bank and the outlet on the bottom as shown in Fig. 2. The outlet-water tubes were in contact with the inlet gas stream, giving in effect counterflow. The screen circuit is shown in Fig. 2, and consists of two staggered rows of tubes above and below the bank, the inlet and outlet rows being the rows nearest the inlet and outlet of the water circuit.

The water temperature in the screen circuit was maintained equal to the water temperature of the bank circuit at the inlet and outlet in order to minimize the radiation effect in the cavity below and above the bank.

The inside of the gas duct between lower and upper screen tubes was lined with polished KA2S plates 1/32 in. thick, which were used because of the low emissivity value of these polished plates, to minimize radiation effect to, and loss from, the side walls.

Tube temperatures were taken by means of iron-constantan

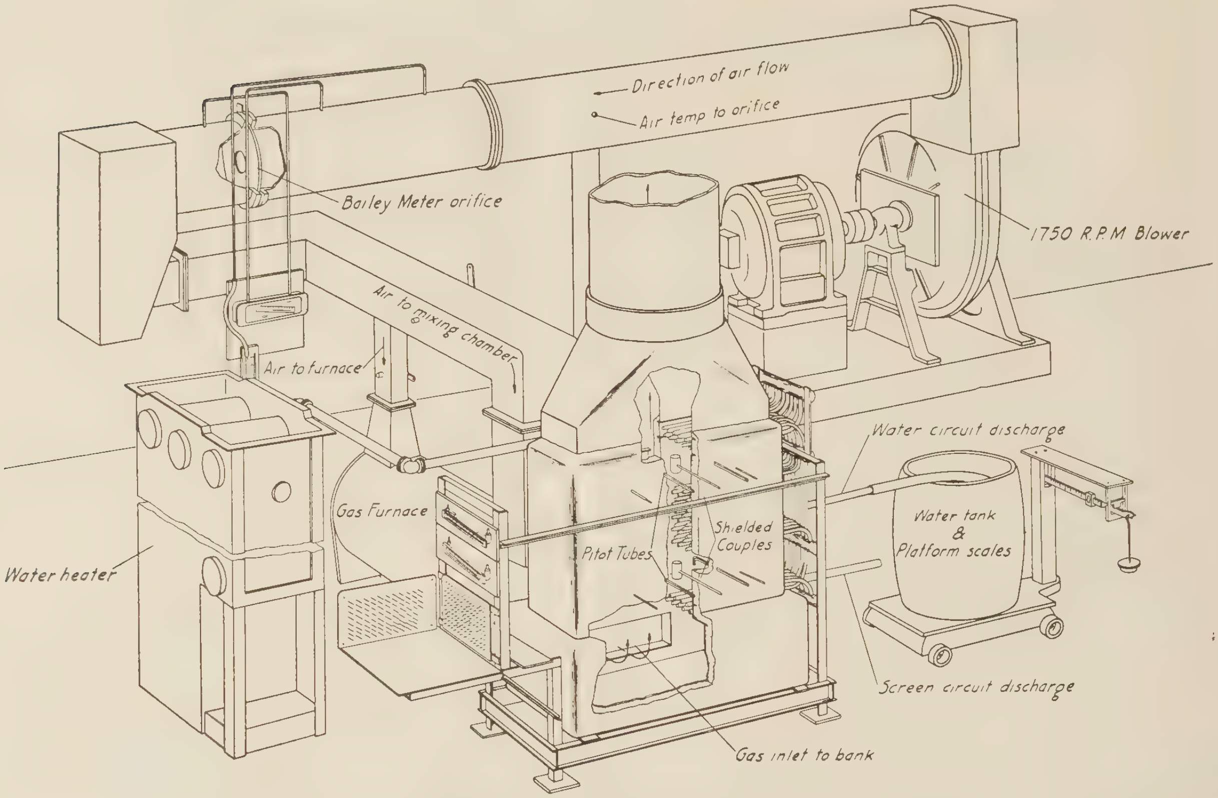

Fig. 2 General Arrangement of Apparatus for Small-Scale Tube Banks

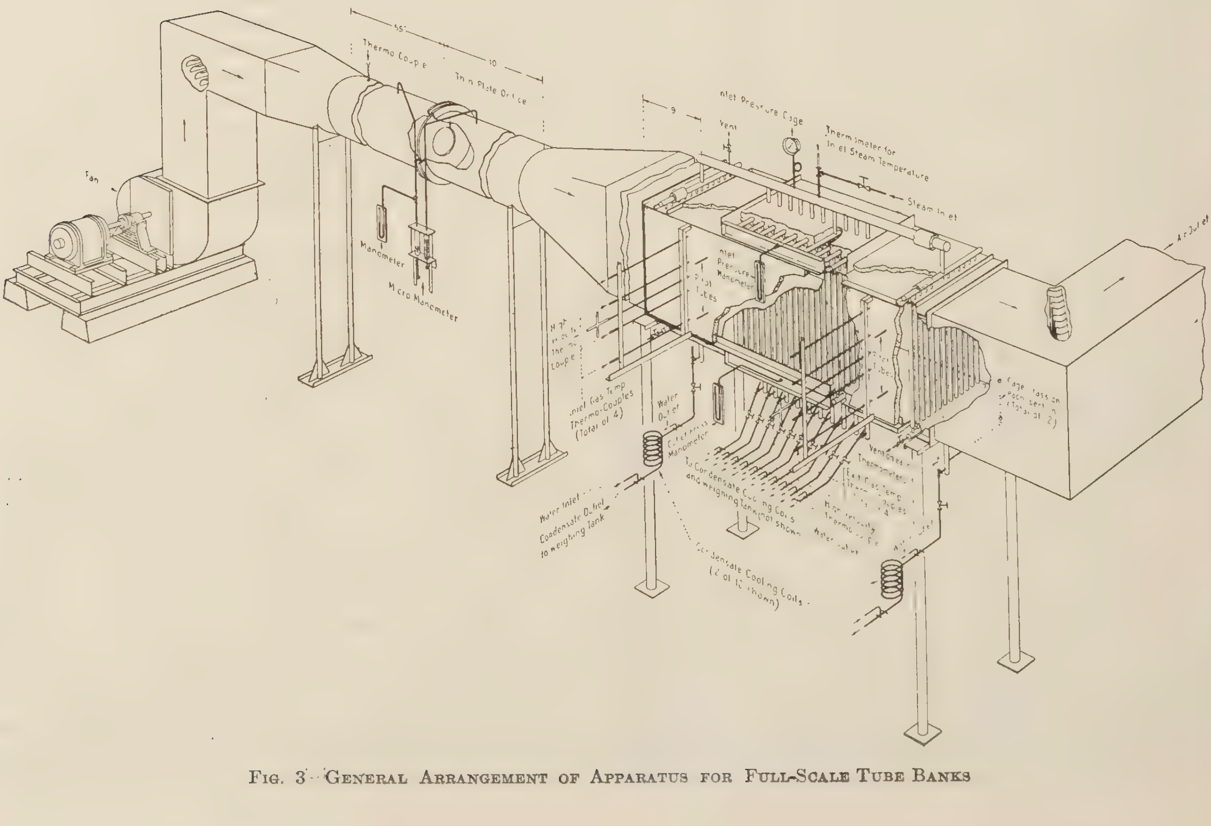

Fig. 3 General Arrangement of Apparatus for Full-Scale Tube Banks

thermocouples peened upon the inlet and outlet tubes of individual elements.

The blower shown in Fig. 2 was directly connected to a 1750-rpm 75-hp motor, and the air flow was regulated by means of dampers in the fan inlet. Three sizes of orifice plates were available to cover the required range of air flow.

The amount of air required at the various points within the apparatus was proportioned by means of conveniently located dampers and the air-gas mixture was proportioned to give a temperature at the bank entrance of approximately 600 F. As previously stated, the water entered both the screen and bank circuit at 90 F with the flow being regulated to give an approximate exit temperature in both circuits of 180 F. The water entrance temperature was maintained at 90 F to eliminate any condensation due to passing through the dew point of the gas on the tube surface. The gas exit temperature was a function of gas and water flow and bank arrangement, and was so governed with an average temperature drop in the gas stream between 300 and 400 F over the range of gas flow.

The testing procedure which was followed was to divide the time devoted to each test into two parts. First, the apparatus was allowed to reach an equilibrium point from a heat-absorption standpoint and then temperature traverses and calibrations, and velocity traverses were made. Second, the top and bottom shielded gas couples and pitot tubes were placed at the duct center line and a complete set of readings was taken. When conditions were found to be stable, at least four sets of observations were recorded from an average of which the results were calculated.

The tube inlet and outlet thermocouples on individual elements were employed as a method of determining the equality of heat absorption across the bank. The temperature increase in the water stream was measured by means of mercury thermometers.

Full-Scale Apparatus. The apparatus for testing the banks of 2-in. tubes is shown in Fig. 3. Room-temperature air was supplied by means of a blower direct-coupled to a direct-current variable-speed motor. The path of the air was through the blower and discharging into a rectangular duct which converged into a circular duct. The circular duct extended approximately 60 ft to a point beyond the orifice where it diverged into a rectangular-shaped duct of dimensions required by the bank of tubes being tested. The rectangular duct housing the bank and radiation screen tubes with a 2 × 3 tube diameter staggered arrangement of 2-in. tubes in place is shown in Fig. 3. The height of the rectangular duct was constant for all arrangements tested, being determined by the height of tube which was 4 ft. The horizontal width of the duct was variable to accommodate various tube arrangements. The inside surface of the duct was lined with transite plates which were faced with cold-drawn polished aluminum plates $^1/_{32}$ in. thick.

The steam circuit of the full-scale apparatus was designed for 450 lb per sq in steam pressure. However, the majority of tests were run at approximately 18 lb per sq in. pressure abs.

The source of steam supply for the apparatus was a gas-fired high-pressure saturated-steam test boiler operated at pressures which, when expanded, gave a slight amount of superheat in the supply nipples to the tubes to insure dry saturated steam. From Fig. 3 it will be noted that the path of the steam was from the boiler to a header serving the nozzles of individual row headers, thence through the individual row headers into the tubes being tested. Condensate at saturation temperature was taken from individual tube rows into their respective cooling coils which discharged into individual-row weighing tanks.

The upper header supplying the individual tube rows and the screen rows was fitted with bleeder lines, one located between inlet screen and first row of the bank, and one between the tenth row of the bank and the outlet screen row.

The radiation screens were composed of a single row of 2-in. × No. 10 Bwg tubes on 3-in. horizontal centers. As previously stated, the inside walls of the duct, with the exception of the upper and lower tube sheets, were lined with $^1/_{32}$-in-thick aluminum plates. The duct was lined with aluminum plate because of the comparatively low emissivity value of the aluminum which helps minimize the radiation to, and loss from, the walls. No further attempt was made to isolate radiation effects nor was any allowance made for the conductance effect of the hot tube sheets, and the side wall adjacent to the bank, upon the gas flowing. Both of the latter effects are considered too minor to have any appreciable effect upon the final results.

For purpose of relieving the steam side of the bank and screen tubes of any entrained air, the individual elements were fitted with $^1/_8$-in. bleeder lines. Due to the higher density of the air than steam at pressures under consideration, the bleeder lines were located on the bottom header of the elements.

Preliminary work with one row of tubes in place indicated that the tube temperatures at the entrance end of the tube were substantially the same as the exit tube temperature and that a $^1/_8$-in. bleeder line was sufficient to air-scavenge the element at all rates. However, in addition to the precaution of installing a single row before proceeding with the ten-row bank, thermocouples were peened on the top and bottom, but within the gas streams, of tubes on No. 1 and No. 10 elements, on the supply nipples to individual row headers and on tube nipples at various locations.

The individual row headers were insulated at top and bottom, as were all steam lines at the top of the apparatus. The condensate discharge lines from the lower individual headers were insulated by means of 2-in. layer of asbestos rope and Superex. The amount of insulation at the top of the apparatus was such that a slight amount of superheat was noticeable in the top tube nipples. The external surfaces of the duct between inlet and exit screen rows were insulated by means of 2 in. of Superex.

The testing procedure of the full-scale apparatus was similar to that of the small scale, i.e., after the apparatus has been allowed to reach equilibrium, the temperature and flow traverses and thermocouple calibrations were made, after which readings were taken. The length of time required for each test was determined by the consistency of the data and the time required to obtain a representative condensate sample. No tests were of less than 30 min duration. The condensate was cooled in coil-type coolers and weighed on platform scales. Complete sets of observations were recorded every 5 min.

It was found necessary to vent the lower headers continuously while operating with low-pressure steam. The quantity of steam vented amounted to 2 or 3 lb per hr per element. It was recognized that a possibility of venting wet steam existed. However, due to the small quantity of steam vented and the low velocities of the bottom header, the condition of excessive loss was considered quite remote. As a further check upon accuracy of results from the foregoing standpoint, the apparatus was operated through flow at ranges of gas flow similar to one or two of the lower rates of gas flow of the dead-end tests. The results of the through flow tests were found at all times to be in agreement with the dead-end tests.

Condensate temperatures were taken at the exit of individual elements by means of mercury thermometers and the similarity of condensate temperatures across the bank was also employed as an indication of air-free condensation. During preliminary operation it was found that a drop in condensate exit temperature was an indication of air binding, which bleeding immediately relieved.

During test operation under through-flow conditions, it was

found that the temperatures indicated by thermometers at element outlets were in excellent agreement with the saturation temperature of steam at the internal element pressure as determined by means of a mercury manometer at the top and bottom of No. 5 element.

Thermocouples and potentiometer were again checked against a calibrated mercury thermometer, all gas couples including the high-velocity couple were checked against each other previous to each test and necessary precautions were taken with gages, connections, pitot tubes, and weighing scales.

The calibration correction of the high-velocity thermocouple was small and its absolute accuracy is probably questionable, but due to the method of its use as well as that of the gas thermocouples the accuracy is well within the allowable range of plus or minus 1 deg.

The guaranteed accuracy of the Bailey Meter Company thin-plate orifice used is 2 per cent. However, judging from the small discrepancies between gas and steam-side data, the actual error is apparently considerably below 2 per cent when an accurately calibrated differential gage is used with the orifice.

The measure of the accuracy of the important measurements again was a heat balance between steam-side dissipation and gas heat absorption. The convection characteristics were determined from steam-side data or condensate weights.

Results

From the observations made during the operation of the apparatus, the results were calculated for reporting in graphical form, for both the small-scale and the full-scale apparatus.

Small-Scale Apparatus. The data of the individual tests were calculated in a straightforward standard manner and it is not believed necessary to enter into a detailed explanation. The quantities of air flow as determined by the thin-plate orifices were corrected to actual flowing conditions. From the temperature of the gas-air mixture, taken ahead of the entrance-screen circuit, the ratio of pounds of mixture to pounds of air at test conditions was determined, or in other words a correction was made for the amount of gas fuel burned which was not metered.

The gas free-flow area for each arrangement was the minimum free-flow area which was used in calculation of gas mass flow.

It will be recalled that the gas temperatures used in calculating the amount of heat released by the gas were taken by means of three shielded thermocouples located at the duct center line, below and above the bank. The shielded-couple gas temperature was corrected for temperature traverse and high-velocity thermocouple calibration.

The mean specific heat of the gas used was that calculated on the basis of a mixture of air and products of combustion of Ohio natural gas with proper correction for water-vapor content of the mixture.

The final results of small-scale tests were calculated on the water-side heat-absorption basis as the water-side data were considered subject to less irregularity than the gas-side data. A heat balance has been prepared for each test which, it is recognized, is not a measure of the over-all accuracy of the test, but is a measure of the accuracy of important measurements. In other words, a factor of primary importance was in closely checking the heat balance between gas dissipation and water absorption before proceeding.

Various small radiation corrections have been applied to the observations taken on the small-scale apparatus.

The net results after applying the radiation corrections is a maximum decrease in gas-side conductance at a low rate of flow (3000 air-mass flow) of approximately 1.5 per cent or a decrease of 0.5 per cent at the high rate of flow (15,000 air-mass flow).

The logarithmic mean temperature difference between the gas side and the water side was determined and, knowing the heat transmitted and the external heating surface, the over-all conductance was determined.

Full-Scale Apparatus. The data of individual tests were calculated in a manner similar to the calculations for the small-scale apparatus and as the calculations are straightforward a detailed explanation would be unnecessary repetition. One exception would be in the calculation of gas-side conductance where a steam-boundary resistance is subtracted from the over-all instead of that of a water boundary or film as in the case of the small-scale apparatus. The conductance of steam was considered constant for all tests and was taken as 2500 Btu per sq ft hr deg F, but the tube temperature is known to be constant and that of the saturated steam within acceptable limits, independent of any such calculation.

Flow-resistance data have been included with all convection tests. Constant-temperature zero heat-transfer or isothermal flow-resistance tests have been run on five of the small-scale tube banks, and three of the full-scale banks, with isothermal runs of one of the small-scale banks at 100, 500, and 700 F.

The method of determining the flow resistance was checked in the full-scale apparatus by use of the impact legs of pitot tubes against static legs which was standard procedure; checking was done by the removal of the bank in the following manner:

A datum point was selected ahead of the entrance screen where static pressures within the duct might be conveniently taken. Static pressures were taken at the datum point with the bank in place over a series of flow rates, then the bank was removed and the duct sealed so that all conditions were similar with the exception of the bank obstruction. The static pressures at the datum point were again taken over a series of flow rates.

The difference between the static heads before and after bank removal should be in agreement with the standard method of determining flow rates.

The foregoing procedure was followed employing a $1^3/_4 \times 2$ tube-diameter in-line arrangement and the results of the three methods of determining draft loss were found in good agreement with each other. Results of tests of the several arrangements and tube sizes have been compared graphically employing a log-log plot of Reynolds' numbers as abscissas and Nusselt numbers and friction factors as ordinates in Fig. 4 to Fig. 8 inclusive and friction factors alone in Fig. 9.

Fig. 4 shows a log-log plot of the results of the $1^3/_4 \times 2$ in-line arrangement of small-scale $^1/_2$-in. and $^{11}/_{16}$-in. tubes and full-scale 2-in. tubes, compared with Pierson's results[3] for the same arrangement of 0.31-in. tubes. A plot of the $1^3/_4 \times 1.82$ in-line data obtained with $^{11}/_{16}$-in. tubes is also included.

It will be noted that one curve represents well the test results for $^1/_2$, $^{11}/_{16}$, and 2-in. tubes. It will also be noted that there is excellent agreement with Pierson's results[3] between $Re = 15{,}000$ and $Re = 50{,}000$ with the Pierson curve assuming a steeper slope below $Re = 15{,}000$.

An inspection of the friction-factor curves indicates good agreement between all factors compared.

Fig. 5 shows a log-log plot of the results of $1^1/_4 \times 1^1/_4$ staggered arrangement for small-scale $^1/_2$-in. and $^{11}/_{16}$-in. tubes and full-scale 2-in. tubes, compared with Pierson's average curves. These figures include low-pressure and high-pressure steam tests on the 2-in. tube apparatus. It will be noted that the small-scale results of this series are in good agreement with Pierson's model results, falling slightly below them in regard to the convection factor for the Nusselt number and slightly above them in regard to flow resistance or friction factor.

Fig. 6 shows a log-log plot of the results of the 2×3 staggered arrangement of the small-scale and full-scale apparatus.

The small-scale and full-scale results of this series are in good

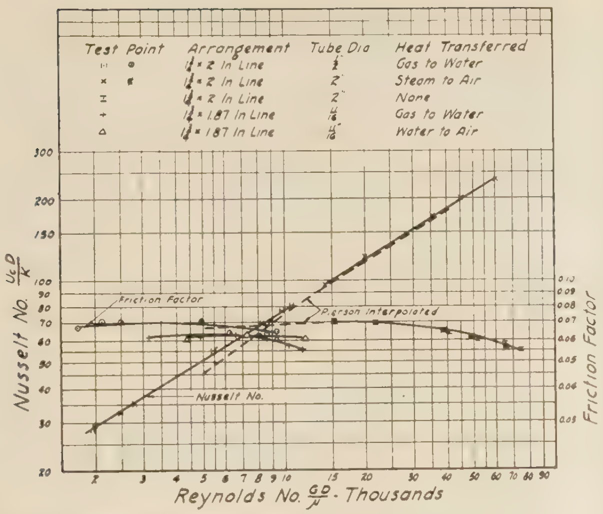

Fig. 4 Results of Tests With Spacings of $1^3/_4 \times 2$ and $1^3/_4 \times 1.82$ Tube Diameters, In-Line

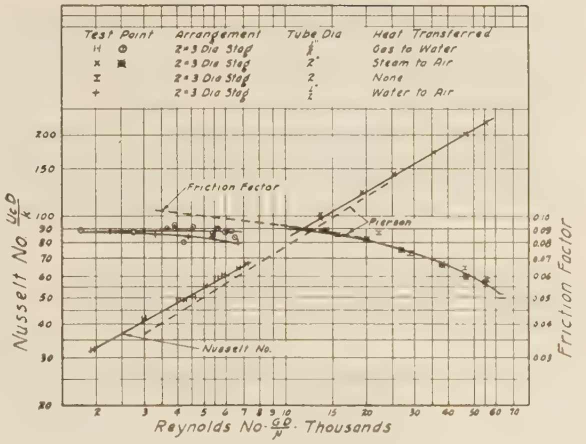

Fig. 6 Results of Tests With Spacings of 2×3 Tube Diameters, Staggered

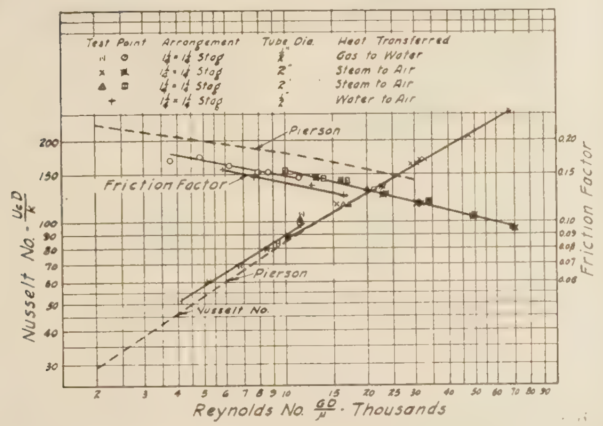

Fig. 5 Results of Tests With Spacings of $1^1/_4 \times 1^1/_4$ Tube Diameters, Staggered

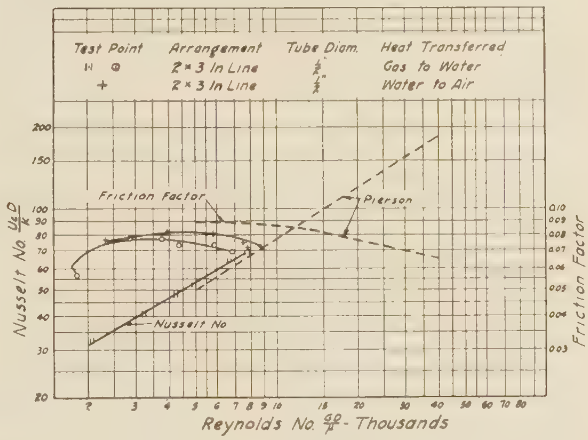

Fig. 7 Results of Tests With Spacings of 2×3 Tube Diameters, In-Line

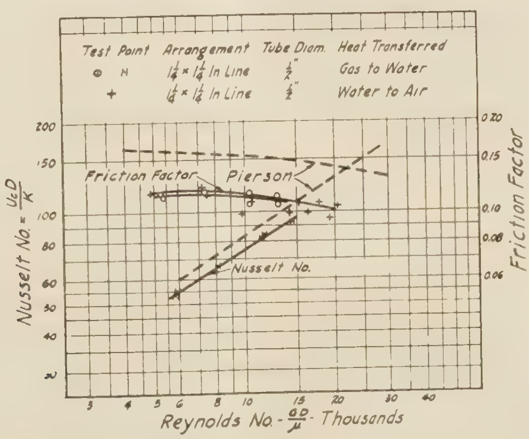

Fig. 8 Results of Tests With Spacings of $1^1/_4 \times 1^1/_4$ Tube Diameters, In-Line

agreement with Pierson's data, again showing a slight disagreement in value of slope with regard to the Nusselt number.

There is good agreement between full-scale and Pierson's data, with the small-scale data of this series falling 10 per cent below Pierson's data.

Fig. 7 shows a log-log-plot of the results of the 2 × 3 in-line arrangement of Pierson's data and those of the small-scale apparatus of this series. Fairly good agreement is indicated for the Nusselt numbers with a slightly greater than usual discrepancy between friction factors. It will be noted that there is good agreement in slope between the two Nusselt relations.

Fig. 8 shows a log-log plot of the results for the $1^1/_4 \times 1^1/_4$ in-line arrangement of Pierson's tests and those for the small-scale of the author's series. The agreement between Nusselt numbers for the two series is fair, but there is a considerable discrepancy between friction factors. The differences between the two series may be due to some dissimilarity, although extreme care was exercised in both cases.

Fig. 9 shows a log-log plot of the constant temperature or zero-heat-transfer air-flow resistance results for the $1^1/_4 \times 1^1/_4$ staggered arrangement of both small-scale and full-scale apparatus. Those for the case when there was heat transfer are shown in Fig. 5. Constant-temperature tests were run on the small-scale apparatus at levels of 100, 500, and 700 F. The constant-temperature tests of the full-scale apparatus were run at room-temperature levels. It will be noted that there is a definite curve indicated for each temperature level tested on the small-scale apparatus with the friction-factor curve from data taken during convection tests falling between the 100-F and the 500-F levels, which is expected.

It will also be noted that the friction-factor curve representing all full-scale results falls somewhere between the family of curves representing the small-scale results.

Generally speaking, it may be said that the agreement between results for small-scale and full-scale apparatus is good with reference to both convection conductance and flow resistance.

The agreement between Pierson's model results[3] and those of the author's results is also considered good with reference to convection conductance, and most of the flow-resistance results fall in line with one or two exceptions.

The difference in slope between the model results[3] and those of the author's results has been more or less apparent throughout the investigation and though no clear-cut explanation has been found, an explanation may lie in the method of evaluating the temperature effect. It will be recalled that tube temperatures in the author's tests were held more nearly constant than in Pierson's work[3] where the electrical input is held constant and tube temperatures are allowed to vary. A more complete discussion of the foregoing is reported by Grimison.[4]

TABLE 2 RATIOS OF TUBE LENGTH TO OUTSIDE DIAMETER OF TUBES FOR PIERSON[3] AND HUGE[1] ARRANGEMENTS

		Huge	
	Pierson	Small-scale	Full-scale
Tube length, in.	9.125	42	48
Outside diameter of tube, in.	0.310	0.502	2
Ratio of tube length to outside diameter.	29.500	83.700	24

Assuming the data and method of correlating results to be correct, the results of Fig. 9 would indicate a definite temperature effect which should be investigated by further tests. It is pointed out that the temperature effect may be a function of tube arrangement and that in order to warrant definite conclusions it may be necessary to test several arrangements.

[4] "Correlation and Utilization of New Data on Flow Resistance and Heat Transfer for Cross Flow of Gases Over Tube Banks," by E. D. Grimison, Trans. A.S.M.E., vol. 59, October, 1937, paper PRO-59-8, pp. 583–594.

It is pointed out here, as shown in Table 2, that the comparisons between small-scale and large-scale arrangements do not satisfy in fact the conditions for exact geometric similarity as the ratio of length to diameter of tube, or space between tubes, varies. It would be difficult to do so and still maintain limits conducive to good accuracy with reference to magnitude of quantities involved.

The similarity in spacing arrangements of the small-scale and full-scale tubes was maintained very rigidly and every effort toward accuracy was expended. Tube sizes, surface conditions, length, and other pertinent factors were carefully checked in an effort to maintain similarity.

Measurements and Observations Made With Small-Scale Apparatus

(*a*) *Air-Flow Data.* (1) The orifice differential was read from a calibrated Ellison gage. (2) The orifice static head was read from an ordinary U-gage. (3) The temperature at the orifice was taken with a mercury thermometer. (4) Wet-bulb and dry-bulb temperatures were read at the blower entrance.

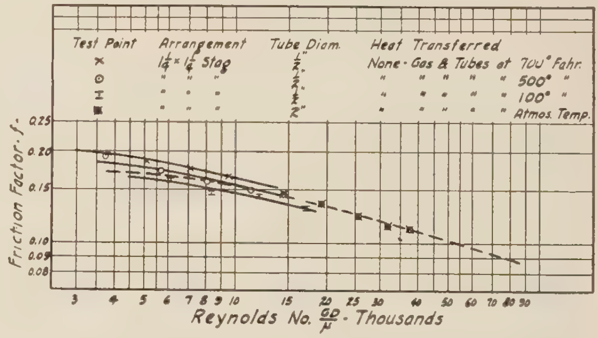

FIG. 9 ISOTHERMAL RESULTS OF TESTS WITH SPACINGS OF $1^1/_4 \times 1^1/_4$ TUBE DIAMETERS, STAGGERED

(5) The barometric pressure was read from a mercury barometer

(*b*) *Gas Data for Gas Leaving the Mixing Chamber.* (1) Adiabatic temperature of the mixture was taken in the duct below the screen tubes. (2) The gas inlet temperature was taken below the bank with bare and shielded couples; the gas outlet temperature was taken above the bank with bare and shielded couples. The gas inlet temperature was maintained at 600 F below the bank. All gas temperatures were taken by means of iron-constantan thermocouples with a standard Leeds & Northrup compensating potentiometer. (3) The temperature traverse across the duct was made with shielded thermocouples at three positions along the duct below and above the bank; the shielded thermocouples were calibrated at one position below and one position above the bank by means of a high-velocity thermocouple. (4) The gas-velocity traverse was made with an A.S.H.V.E. pitot tube below and above the bank. (5) The gas-flow resistance across the bank was measured by means of static legs of pitot tubes; the static-leg results were also checked by means of an impact tube. Both static flow and impact-flow resistance traverses were made west to east across the duct. (6) The shielded and high-velocity thermocouples and potentiometer used were calibrated against a Bureau of Standards certified mercury thermometer before and after each test series. (7) The duct temperature was taken by means of iron-constantan thermocouples imbedded in the duct wall.

(*c*) *Water-Flow Rate.* (1) The water was measured continuously in a tank weighed on platform scales calibrated before and after each test series. (2) The elements were individu-

ally calibrated and checked for equality of flow before installation in the duct; the inlet to the elements was from a common header, while the outlet from the elements was also to a common header.

(*d*) *Water Temperatures.* (1) The inlet and outlet temperatures of both screen and bank circuit were taken by means of twenty 220-F mercury thermometers calibrated before and after each test series. The thermometers were read to 0.2 F. The average inlet water temperature was maintained at 90 F while the average outlet water temperature was maintained at 180 F.

Measurements and Observations Made With Full-Scale Apparatus

(*a*) *Air Flow.* (1) The orifice differential was read with a micromanometer gage. (2) The orifice static head was read with a U-gage. (3) The temperature was read ahead of the orifice. (4) The wet-bulb and dry-bulb temperatures were read at the blower entrance. (5) The barometric pressure was read with a mercury-column barometer.

(*b*) *Air Temperature.* (1) The temperature of air entering and leaving the bank was read with iron-constantan thermocouples. (2) The temperature traverse was made at the entrance and exit of the tube bank. (3) A high-velocity-thermocouple traverse calibration was made at the entrance and exit of the tube bank.

(*c*) *Pressure Drop.* (1) The pressure drop was measured by means of the static leg of pitot tubes; four tubes were evenly spaced across the entrance and four tubes were spaced across the exit flow areas with minimum of ten tube diameters between the measuring instruments and the upstream tube rows. The duct of the apparatus was checked in the following four positions along its length for any irregularities in flow: Ahead of orifice; between the front radiation screen and bank inlet; between the bank outlet and rear radiation screen; and ahead of the 90-deg straightening vanes at the discharge end. These checks were made by means of pitot-tube traverses with room-temperature air flowing.

(*d*) *Steam Pressure.* (1) The steam pressure at the header to individual row header was measured by mercury manometers. (2) The steam pressure at the bottom of individual row header was measured by a mercury manometer.

(*e*) *Steam Temperature.* (1) The condensate temperature was read at the discharge end of the individual rows. The steam temperature corresponded to the saturation temperature of average pressure in tubes.

(*f*) *Condensate Level.* The condensate level was maintained below the lower header by means of individual gage glasses for each element. The level was held at the same elevation at beginning and end of test.

(*g*) *Condensate Weights.* The discharge from the individual-row headers was measured in tanks weighed on platform scales.

Correlation and Utilization of New Data on Flow Resistance and Heat Transfer for Cross Flow of Gases Over Tube Banks

By E. D. Grimison,[1] New York, N. Y.

In spite of the commercial importance of the tube bank in heat-transfer equipment, few experimental data previously have been available on convection heat transfer and resistance to flow of gases sweeping the tubes in cross flow. The new experimental observations reported in the research program[2] of the Babcock & Wilcox Company by Huge[3] and Pierson[4] are therefore of the utmost importance. Results of tests are reported for a large number of tube spacings of the commonly used staggered and in-line arrangements with different tube sizes, with a sufficient number of checks to assure their general accuracy. Special equipment was built and tested by Huge[3] to investigate the influence of equipment size on convection and flow resistance and the results appear to confirm substantially the validity of the similarity principle.

Before these new data could be used in practice, a great deal of analytical study has been necessary; this is the subject of the present paper. It was considered essential to satisfy the following major requirements:

(*a*) Develop or select a method of correlation and of application of the original data to commercial equipment, for which conditions of size and operation may differ greatly from those of the experiments. The general relations developed from the principle of similarity, which have been used by Pierson[4] and Huge[3] in presenting their results, were selected for this purpose initially.

(*b*) Examine the general relations for validity. These appear to be satisfactory except that a few uncertainties exist in details of application.

(*c*) Study differences in characteristics from available sources from the viewpoint of causes and where necessary modify the original data consistently to reconcile such differences.

(*d*) By fairing methods correct accidental errors of small magnitude.

(*e*) Define the relation of the data to the whole problem or process and indicate the method of application to working needs in a sound general procedure.

(*f*) Compare the over-all calculated results with results of reliable field tests of commercial equipment, as well as with experimental results of other investigators. These comparisons indicate that the new data are satisfactory with regard to both accuracy and utility.

All of these phases are discussed at some length in this paper, and the results are summarized in the form of working relations and charts.

1 INTRODUCTION

General Relations

As some of the terms used, and the theoretical background of this development are not familiar to some engineers who have not been concerned directly with analytical work in heat transfer and flow resistance, a brief review of this phase is given.

The bulk of the data used in this paper represent results of tests on banks of 0.3-in. diameter electrically heated tubes, cross-swept by air in turbulent flow. For application of these experimental data to commercial equipment such as the various tube banks of the power boiler, where size and operating conditions are so greatly different from the experiments, a well-founded theory is essential.

For both convection and flow resistance, theoretically developed general relations were available initially, having a common background. In both instances, experimental confirmations of the theory have been obtained for certain types of flow and surface arrangement, but no satisfactory confirmations are on record in the literature for the cross-swept tube bank. The experimental investigations reported by Huge[3] and Pierson[4] furnish substantial confirmation of the general relations, although some new uncertainties in details of application have been revealed. These are discussed later, and for the present the theory will be reviewed briefly without reference to these questions of application. Rigorous treatment becomes complicated and

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[2] The Babcock & Wilcox Company's research program on convection heat transfer and flow resistance in cross flow of gases over tube banks includes (1) the experimental investigation of the influence of tube arrangement, (2) experimental investigation of effects of equipment size, and (3) correlation and utilization of new data regarding this subject. All three phases of the program are reported in this issue of the A.S.M.E. Transactions; the first by O. L. Pierson, the second by E. C. Huge, and the third by the author. See footnotes 3 and 4 on this page.

[3] "Experimental Investigation of Effects of Equipment Size on Convection Heat Transfer and Flow Resistance in Cross Flow of Gases Over Tube Banks," by E. C. Huge, Trans. A.S.M.E., vol. 59, October, 1937, paper PRO-59-7, pp. 573–581.

[4] "Experimental Investigation of the Influence of Tube Arrangement on Convection Heat Transfer and Flow Resistance in Cross Flow of Gases Over Tube Banks," by O. L. Pierson, Trans. A.S.M.E., vol. 59, October, 1937, paper PRO-59-6, pp. 563–572.

Contributed by the Heat-Transfer Committee of the Process Industries Division for presentation at the Annual Meeting of The American Society of Mechanical Engineers, to be held in New York, N. Y., December 6–10, 1937.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1937, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

has been avoided here, but more extended and more or less rigorous developments are readily available (3, 4, 5, 6).[5]

For both convection heat transfer and resistance to the flow of the fluid which is responsible for the heat transfer, the theoretical developments are based on the principle of similarity. This principle is a statement, of that class known in mathematics as "self-evident truths," to the effect that for two fluid streams to be similar, it is necessary that all velocities in one of the streams be in constant proportion to the velocities at corresponding points in the other stream, and in the same directions, and likewise that viscosities, densities, and other significant characteristics must be in constant ratio at all points. It is, of course, necessary that the solid boundaries of the stream or the submerged bodies swept by it be geometrically similar, in regard to both arrangement and surface condition, although there are exceptions to the latter.

By suitable mathematical treatment, certain criteria of similarity of fluid streams are developed, and are properly related. One of these criteria, the Reynolds number, has become quite generally known and accepted and is a measure of dynamic similarity. The Reynolds number consists of four variables arranged as a ratio; viz., velocity V and weight density ρ of the fluid, a characteristic linear dimension of the solid boundary such as a diameter D, and absolute viscosity of the fluid μ, in the form Reynolds' number $Re = \rho VD/\mu$ which is dimensionless. All four of these variables may differ in two fluid streams, but if the values of the Reynolds numbers for the two streams are the same, they are dynamically similar by definition.

Theoretically, certain flow-resistance effects are the same for two dynamically similar streams within similar channels. Convection effects are similar if the streams are both dynamically and thermally similar. Thermal similarity is defined by another group of variables. Without extending the discussion of the nature of these criteria, the general relations resulting from the theoretical treatment may be given and defined. Not all authorities use the same forms, but those which have been adopted in this paper are probably most generally used in this country.

For convection heat transfer, the general relation has the form of Equation [1], in which a whole group of variables is treated as a single variable, the group having zero dimensions. For a single gas, the second group on the right side is constant with respect to temperature, and as all of the data with which these reports are concerned are for air, the general relation is simplified to the form of Equation [2], which is more convenient and is adopted for use. Transposition leads to Equation [2a], giving convection conductance, or heat-transfer coefficient U_c at the gas boundary or film. These Equations are

$$\frac{U_c D_t}{k} = A \left(\frac{GD_t}{\mu}\right)^m \left(\frac{c_p \mu}{k}\right)^n \qquad [1]$$

or, for a single gas

$$\frac{U_c D_t}{k} = B \left(\frac{GD_t}{\mu}\right)^m \qquad [2]$$

$$U_c = B \left(\frac{k}{\mu^m}\right) \frac{G^m}{D_t^{1-m}} \qquad [2a]$$

where U_c = convection conductance at solid boundary, Btu/[(hr)(sq ft)(deg F)]; D_t = tube diameter, ft; k = thermal conductivity of gas, Btu/[(hr)(ft)(deg F/ft)]; μ = absolute viscosity of gas, lb/(hr)(sq ft); $G = \rho V$ = mass (or weight) flow of gas, lb/(hr)(sq ft); ρ = weight density of gas, lb per cu ft; V = velocity of gas, ft per hr; c_p = specific heat of gas at constant pressure Btu/(lb)(F) A, B, m, and n = values to be determined by experiment, dimensionless; $\frac{U_c D_t}{K}$ = Nusselt (or convection) number, dimensionless; $\frac{GD_t}{\mu}$ = Reynolds' or flow number, dimensionless and $\frac{c_p \mu}{k}$ = Prandtl (or fluid-properties) number, dimensionless.

In general terms, Equation [1] means that for two similar fluid streams with heat transfer between the fluid and its solid boundaries, the Nusselt numbers standing for convection effects must be the same and must be functionally related to the Reynolds and Prandtl numbers, the first being the criterion of dynamic, and the two together of thermal similarity.

The fundamental general relation for flow resistance which also groups the variables in dimensionless ratios, is omitted here, to save detailed explanations. The general relation of Equation [3], which is used in the present work, is derived from it after modifications permitting the use of convenient units for the several variables, and is sound to the same degree as the fundamental relation. This equation is

$$\Delta P = \frac{fNG^2}{10.84 \times 10^8 \rho} \qquad [3]$$

where ΔP = total pressure drop across bank, in. of water; f = a friction factor which is a function of Reynolds' number, dimensionless; N = number of major restrictions encountered in flow through the bank; and 10.84×10^8 = a constant which includes the acceleration of gravity in ft per hr per hr and the conversion from lb per sq ft to in. of water, lb/[(ft)(hr^2)(in. water)]. The friction factor

$$f = \text{function } (GD_t/\mu) \qquad [4]$$

The relation of Equation [3] is of the general Fanning type, and treats the pressure drop across the bank as being due to flow through a series of orifices formed by the successive major restrictions in the flow paths.

The effects of mass-flow rate, of size, and of fluid temperature are automatically accounted for in these relations if the theory is correct, so that in comparing convection in two tube banks of different arrangements, the ratio of Nusselt numbers is identical with the ratio of convection heat-transfer rates, for equal mass flow, temperatures, and tube sizes in the two cases. Similarly, the ratio of values of friction factors is identical with the ratio of pressure drops for two arrangements, under the same qualifications. It should be noted that the equalities of these ratios apply to equal mass-flow rates in pound per hour per square foot of flow area, and not to equal loads in pounds of gas per hour per square foot of surface.

Validity of General Relations

The practical objective of formal correlation of experimental data is to facilitate their safe extension to conditions of useful applications. Completeness of the selected general relations and satisfactory correlation within experimental ranges are, of course, essential. Huge[3] and Pierson[4] have presented their data on heat transfer in plots of Nusselt versus Reynolds' number representing Equation [2], and on flow resistance as plots of friction factor versus Reynolds' number representing Equation [4]. These are probably the best initial choices as bases of correlation, but are subject to examination for completeness and validity, and proper procedure in details of application must be developed empirically from the test results.

From the engineering viewpoint, the validity of Equations [2], [3], and [4] is best considered in several phases; i.e., in relation

[5] Numbers in parentheses refer to the Bibliography at the end of the paper.

to the influences of rate of gas flow, of equipment size, and of gas temperature and chemical composition or physical properties. It is of course clear initially that the numerical values of proportionality factors and perhaps of exponents also will vary with different arrangements or spacings of tubes in a cross-swept bank, and identification of these variations was the primary objective of investigation. Mass flow G is used in the general relations, rather than the product of velocity and specific weight or mass density which are sometimes evaluated at different temperatures. The procedure used in this paper is common practice in this country and is supported by Pierson's results[4] for varying air densities. It had previously been well-established for cases other than the cross-swept tube bank.

Of paramount importance with reference to application, is the validity of the relations in respect to size, as represented by tube diameter in Equations [2] and [4]. This appears to have been confirmed in the present investigation beyond reasonable doubt, in the range of 0.3-in. to 2-in. tube diameters, as reported by Huge.[3] Considering the departures in many details from exact similarity in tests of the several sizes, this confirmation is remarkably good, and application of the results to long tubes of diameters up to 4 in. and perhaps larger would appear to be entirely safe.

In later detailed consideration of the data, certain modifications are made in the results of tests by Pierson[4] on the 0.3-in. tubes in respect to convection characteristics only, from comparisons with Huge's results[3] for larger tube sizes. Although the causes of the qualitative discrepancies leading to this general modification of Pierson's convection data[4] have not been discovered, it is unlikely that disparity in size is responsible. The disparity is more probably connected with the situation as to temperature effects, which for convection especially are not clearly established. The influence of temperature is the third feature of practical concern, and is included in the general relations indirectly, through the fluid physical properties of thermal conductivity, absolute viscosity, and density. All of these vary with temperature, and a theoretical treatment has not been established. This leads to considerable confusion in practice, and several procedures have developed without satisfactory experimental confirmation in any case.

2 CORRELATION OF DATA ON FLOW RESISTANCE

Although the present investigation was undertaken primarily to investigate the influence of tube arrangement on thermal performance of tube banks cross-swept by gases, the measurements of flow resistance have become the feature of probably greater interest, both practical and scientific. Great variations in flow resistance have been found, particularly for tubes arranged in-line, accompanied by relatively small changes in convection conductance.

Huge[3] and Pierson[4] show the results of their measurements as values of the friction factor f in the general relation of Equation [3]. As indicated by Equation [4], f would be expected to vary as a function of Reynolds' number, without regard to values of the individual members of the latter. This is supported by a wealth of data on flow in pipes and through orifices, and within limits has been shown for cross flow over tube banks also, the latter data having been summarized by Chilton (7). For most of these cases, in the turbulent-flow region, Equation [5] is valid over a wide range of Reynolds' numbers, the proportionality factor C and exponent X varying with factors of geometry of the flow channel and solid boundaries, that is

$$f = C\left(\frac{GD}{\mu}\right)^X \qquad [5]$$

Before any conclusions were reached concerning the validity of Equation [5] for application to tube banks, it became necessary to investigate empirically the influence of gas temperature on the flow resistance or friction factor f. The conventional practice of evaluating gas density at mean fluid temperature was clearly not satisfactory. Empiric analyses were made of variations in flow resistance from full and reduced electric-power input, and isothermal runs by Pierson,[4] and of isothermal and nonisothermal tests with (a) gas cooling, and (b) gas heating by Huge[3] on his small-size apparatus. The definitions of Equations [6], of the significant gas temperature for evaluation of the fluid density (or specific volume), were indicated by this study, and permitted satisfactory correlations to be made of these two groups of data; for staggered arrangements

$$t_d = t_t \pm 0.8\ \Delta t_m \qquad [6a]$$

and for in-line arrangements

$$t_d = t_t \pm 0.9\ \Delta t_m \qquad [6b]$$

where

t_d = temperature for evaluation of gas density in Equation [3], F; t_t = mean tube temperature, F; and Δt_m = mean temperature difference between gas and tubes, F. The sign of the second term on the right, is plus for gas cooling by heat transfer to cool tubes, and minus for the opposite direction of heat flow.

This procedure was later applied to results of tests by Huge[3] on banks of 2-in. tubes when these became available, the data including nonisothermal runs at two different tube temperatures in the case of one arrangement, and isothermal runs on all three. Correlations of flow-resistance results on this basis were excellent.

The validity of the general relation is clearly indicated by the comparisons of tests on large and small tubes shown by Huge.[3] The differences between results on the several test setups for the $1^1/_4 \times 1^1/_4$ staggered banks are regrettably large, but probably are due to the great difficulties in making accurate measurements on such closely spaced tubes, where the free space is only one quarter of one tube diameter, rather than to the invalidity of the general relations. The large effect of small errors in end clearances on the indicated values of the friction factors has been shown by Pierson,[4] who had to follow a very painstaking procedure in setting up his closely spaced banks of small tubes.

The comparisons of results for the $1^1/_4 \times 1^1/_4$ arrangements, both staggered and in-line, show Pierson's values[4] of friction factor to be characteristically higher at close spacing than Huge's.[3] In the absence of any discovered evidence of error in the results of either investigator, there is no choice but to average the results at the close spacings. This practice is supported by results of tests on 2-in. tubes.

Averaging was accomplished by plotting and drawing mean curves representing the results generally. This was necessary because the data from the small-sized banks do not cover the same range of Reynolds' numbers. In this connection, attention is called to the characteristic shapes of the curves of friction factors versus Reynolds' numbers for the small-sized apparatus. In most cases, at the lower end of the range of Reynolds' numbers, the friction-factor curve tends to hook rather sharply downward. When several sizes of tubes were tested, so that the Reynolds-number ranges of the several test series overlapped, this downward hook intersects the horizontal portion of the characteristic curve for the next smaller size. It appears fairly evident that the general characteristic curve is a nearly horizontal line at lower Reynolds' numbers, inclining downward with Reynolds' numbers, above 10,000. This general shape has been followed in most cases, averaging the original values, with exceptions in a

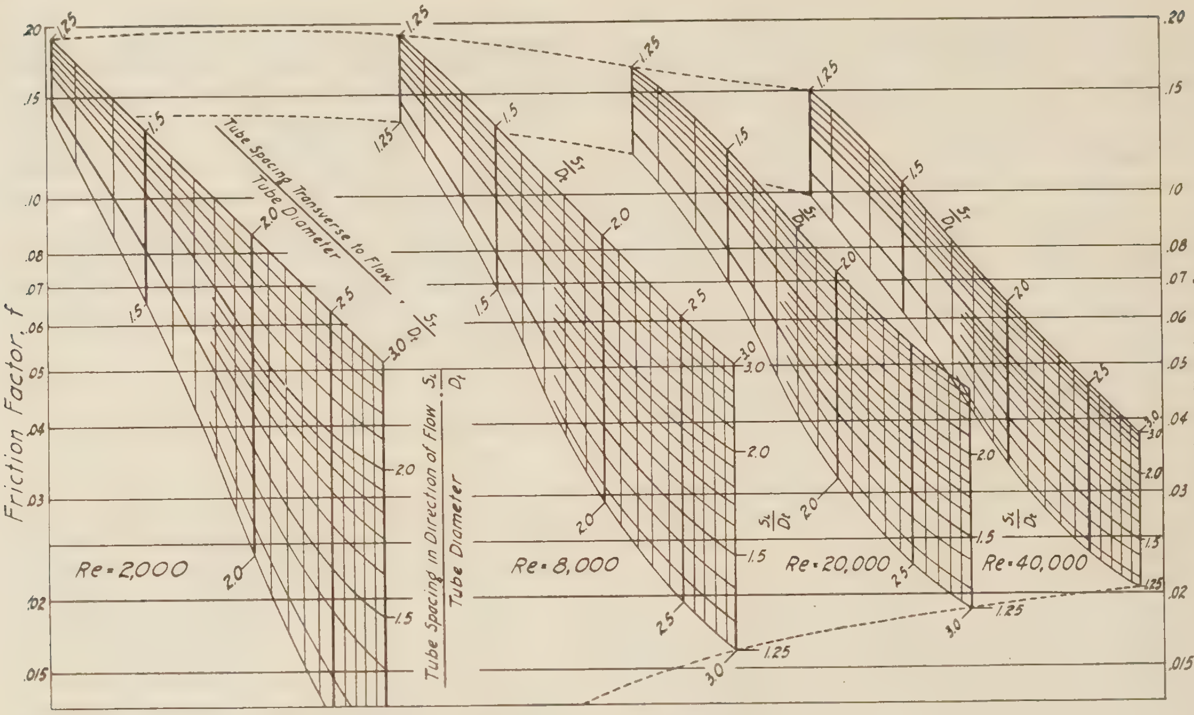

FIG. 1 FRICTION FACTORS FOR IN-LINE TUBE BANKS

few instances in which the friction-factor curves are generally U-shaped or in which the friction factor increases with the Reynolds number.

After slight modifications of the friction-factor values in line with the preceding observations, values were plotted and cross-plotted in relation to transverse and longitudinal spacings at several values of the Reynolds number, including $Re = 2000$ to which value the original results were extrapolated where necessary. Values from tests of 2-in. tube banks, which were considered most reliable, and from average curves of the Pierson[4] and Huge[3] results, were used as guide points in plotting and fairing. In effect, this procedure introduces generally minor modifications to the original Pierson data,[4] bringing them consistently into line with the large size and average values which were accepted as reliable.

The large number of data which have resulted from this investigation are conveniently summarized in three groups, as follows: (*a*) In-line arrangements, in which the number of major restrictions N equals the number of rows. (*b*) Staggered arrangements, with minimum flow area in transverse intertube spaces, or openings, and N equal to the number of rows. (*c*) Staggered arrangements, with minimum flow area in diagonal intertube spaces or openings between tubes, and N equal to number of rows less one.

The friction factor values for these three groups of arrangements are summarized in the three-dimensional charts shown in Figs. 1, 2, and 3. These charts represent in effect a combination of four conventional charts for $Re = 2000$, 8000, 20,000, and 40,000, arranged to be viewed in perspective, and showing the friction factor f in relation to transverse and longitudinal center-line spacing in terms of tube diameters, or S_t/D_t and S_l/D_t, respectively. If a curve is drawn through the friction-factor values of the four fields for a given arrangement, the f–Re characteristics, similar to those shown by Huge[3] and Pierson,[4] result. The Reynolds-number scale is of course not fixed.

APPLICATION TO CALCULATIONS

The general relation of Equation [3] should be directly applicable to calculations of draft loss or pressure drop, with of course due allowances for stack effect and differences in velocity head between gage-connection points wherever these corrections are significant.

For practical purposes, sufficient accuracy is had by taking from Figs. 1, 2, or 3 the values of f at Reynolds' numbers spanning the one in question, and interpolating arithmetically to determine the value sought. The gas density in Equation [3] should be evaluated at a temperature defined by Equation [6], but approximate methods of handling this sometimes troublesome detail are easily developed for various special conditions of application.

Entrance and exit losses are included in the values of the friction factor, which therefore apply strictly only to a tube bank ten rows deep. For banks of less depth, correction factors may be taken from Fig. 17 of Pierson's paper,[4] showing the results of direct tests on two banks, one staggered and one in-line, of which the depths were varied systematically. As Pierson's data[4] apply to specific tube spacings, they are not exact for others, but probably the corrections are sufficiently accurate for general practical use. For banks deeper than ten rows, the corrections are small enough to be neglected.

It should be noted particularly that these values of friction factor correspond to the use in Equation [3] of values of mass flow G for the minimum flow area between tubes of the bank, whether this occurs in transverse or in diagonal openings. The definition of N in Equation [3] as the number of minimum restrictions must also be carefully regarded.

ACCURACY OF RESULTS

The general accuracy of these data is of high order, probably much better than can be expected from the run of field-test results. Certain modifications to the original data of Pierson[4]

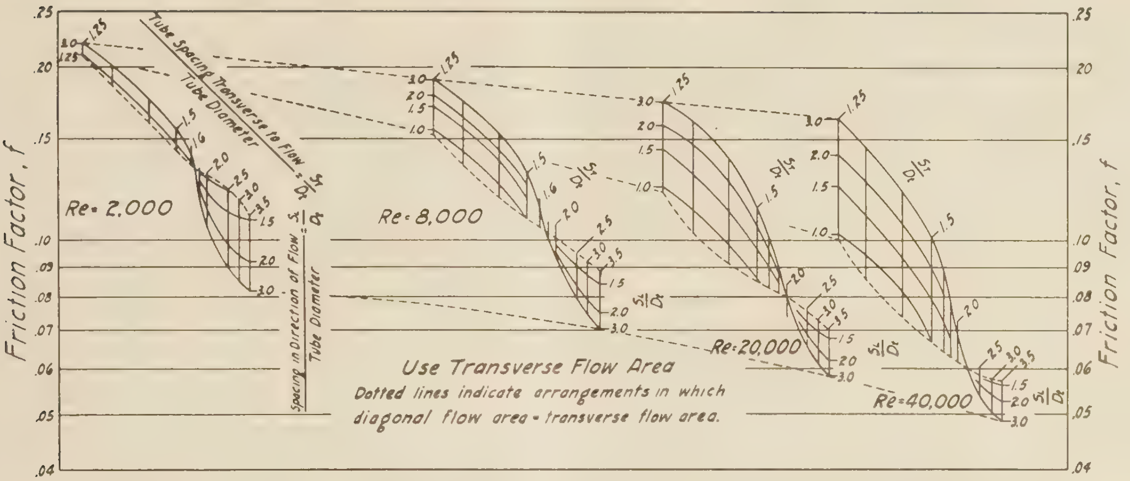

FIG. 2 FRICTION FACTORS FOR STAGGERED-TUBE BANKS WITH MINIMUM GAS-FLOW AREA IN TRANSVERSE OPENINGS

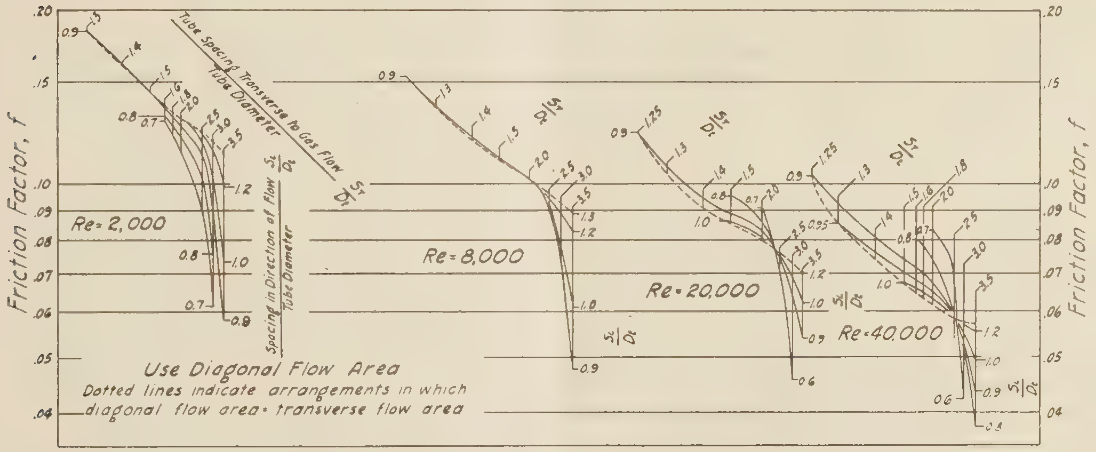

FIG. 3 FRICTION FACTORS FOR STAGGERED-TUBE BANKS WITH MINIMUM GAS-FLOW AREA IN DIAGONAL OPENINGS

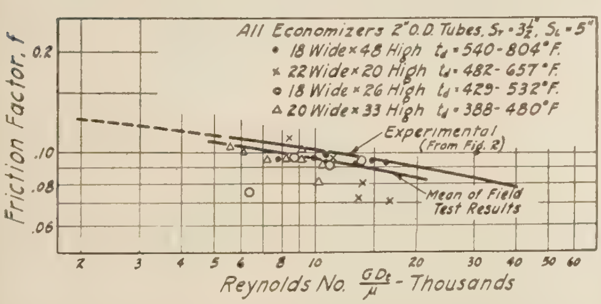

FIG. 4 COMPARISON OF FRICTION FACTORS WITH FIELD MEASUREMENTS OF DRAFT LOSSES IN ECONOMIZERS WITH STAGGERED-TUBE BANKS

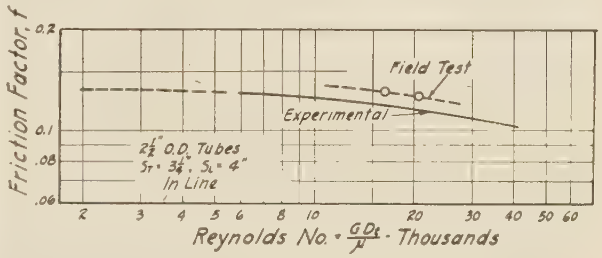

FIG. 5 COMPARISON OF FRICTION FACTORS WITH FIELD MEASUREMENTS OF DRAFT LOSSES IN A CROSS-FLOW AIR HEATER WITH IN-LINE TUBE BANK

and Huge[3] have been found necessary, but these apparently are in the direction of improvement in accuracy. Comparisons of the final data with the original values have been omitted here because of space requirements, but in general changes have been small, affecting shape of curves rather than position, with some exceptions. For the bulk of the data, the maximum error is of the order of ±5 per cent and the average is smaller. The limits of accuracy are somewhat greater for staggered banks in which the diagonal flow area is the minimum.

The direct practical utility of these data may be judged by comparisons with suitable field-test results. Although a great mass of field data are available, only a very few reports are suitable for this purpose. Most of them must be discarded because of the occurrence of bends and changes in section between measuring points, of lanes within the bank or at the sides, or because the locations of instrumental connections were not reported or could not be determined with any certainty.

Economizer field tests were found useful for purposes of comparison, all representing a $1^3/_4 \times 2^1/_2$ staggered arrange-

ment of 2-in. tubes. Friction factors from some of these field tests are plotted in Fig. 4, and compared with a curve of values from Fig. 2 for this arrangement. The mean curve of the field data is about 5 per cent lower than that of the experimental values. Of greater significance in view of the high values of f, for closely spaced tubes, is the comparison in Fig. 5 of values of friction factors from Fig. 1 for a 1.3 $\times$ 1.6 in-line bank of $2^1/_2$ in. tubes, with recent results of careful field measurements. Here the field results are about 7 per cent higher than the experimental values, corresponding for instance to a calculated value of 4 in. of water and a measured value of 4.3.

Comparisons With Previous Experiments

Data of previous experimental investigations, covering a period of years and undoubtedly of variant accuracy, have been collected and recalculated by Chilton (7) on a basis substantially the same as that used here. Chilton uses the clearance between tubes as the characteristic dimension in Equation [4] rather than the tube diameter, but the conversion is easily made to the basis used here.

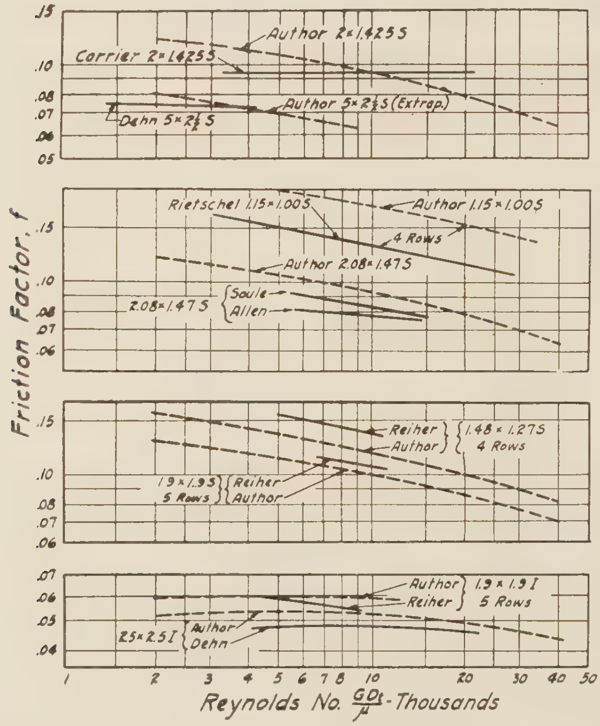

Fig. 6 Comparison of Friction Factors With Experimental Results of Other Investigators as Given by Chilton (7)

Comparisons of the data shown graphically in Figs. 1, 2, and 3 with values of friction factors as given by Chilton are shown in relation to Reynolds' number in Figs. 6 and 7. Tube spacings are indicated on the curves in terms of tube diameters; for example, (2 $\times$ 1.4S) indicates a staggered tube arrangement S spaced at two tube diameters transversely, and at 1.4 diameters longitudinally, of direction of fluid flow. In general, the agreements are more numerous than were expected from previous acquaintance with the data of the literature. Of particular interest are Sieder's data (8) for water, kerosene, light and heavy oils, representing very extensive tests on a bank of tubes spaced at 1.25 $\times$ 1.08 diameters, staggered. The exact agreement of Sieder's data (8) on a variety of liquids with the data shown graphically in Fig. 3 for air is important in several respects. It confirms the validity of Equations [3] and [4] for all fluids, as well as the relatively high values of the friction factors which have been given for close tube spacing. Chilton improved Sieder's correlation of his data by use of an empirically determined temperature basis of evaluating fluid density. The definitions given by Chilton of the significant fluid temperature are similar to those of Equation [6] except that the numerical factor in the second term on the right-hand side is given as 0.8 for fluid heating, and 0.68 for fluid cooling. The first value is the same as that of Equation [6a], applying to staggered banks. Presumably all of Chilton's data are evaluated on this basis, and therefore are comparable with those reported here.

An extensive experimental investigation of flow resistance of cross-swept tube banks was reported by Brandt (9, 10). Wooden cylinders were arranged in banks six rows deep and of varying widths, between glass end plates; a few tests were made using iron tubes. Diameters were 39.8, 59.6, and 79.8 mm for wooden, and 63.6 mm for iron tubes, which for a fixed tube length of 100 mm, correspond to length-diameter ratios from 2.52 to 1.25; extremely small values in all cases. Original data of a great many tests on a wide variety of arrangements are not given, and comparisons have been made with values calculated from Brandt's general relation and accompanying graphs of constants. These comparisons are too extensive for inclusion in this paper, but large discrepancies were found. The new values range from 54 per cent higher to 22 per cent lower than Brandt's for corresponding arrangements in-line, with one or two wider discrepancies and from plus 50 per cent to minus 15 per cent for staggered banks. The discrepancies do not vary systematically with tube arrangement.

3 CORRELATION OF DATA ON CONVECTION HEAT TRANSFER

Experimental data on convection heat transfer have been reported by Pierson[4] and Huge[3] as plots (on logarithmic coordi-

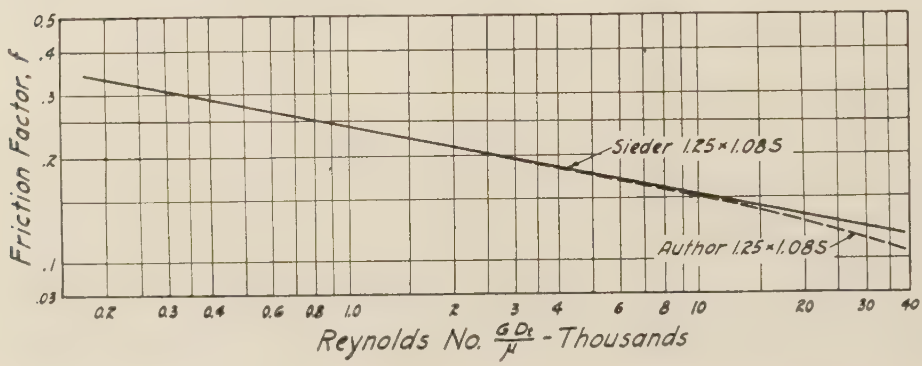

Fig. 7 Comparison of Friction Factors With Sieder's (8) Data for Liquids as Modified by Chilton (7)

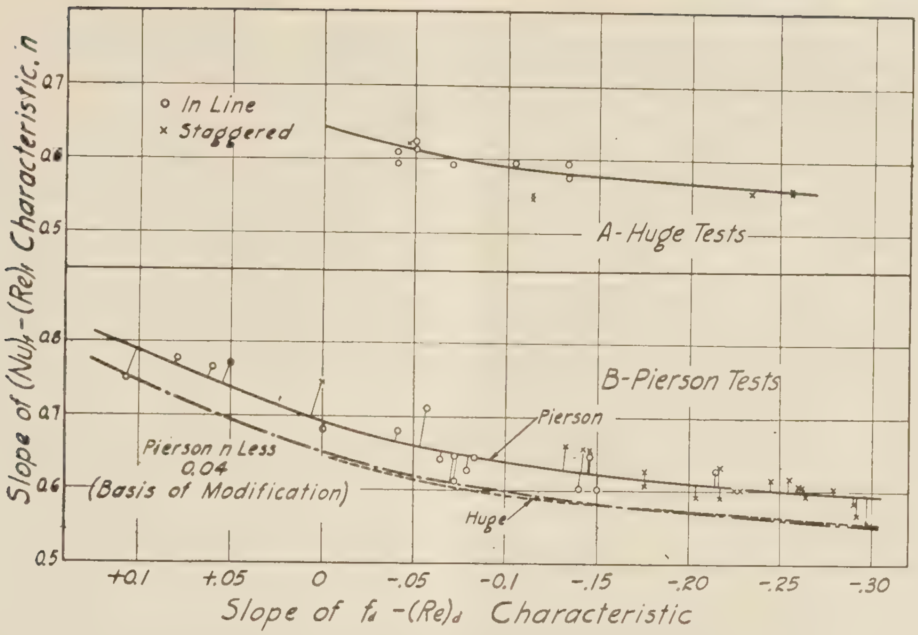

Fig. 8 Relation of Slopes of Curves of Nusselt Numbers Versus Reynolds' Numbers to Approximate Average Slopes of Curves of Friction Factors Versus Reynolds' Numbers

nates) of Nusselt numbers in relation to Reynolds' numbers, these terms having been defined and discussed previously. In Figs. 4 to 9, inclusive, of Huge's paper, his results are compared with Pierson's on several similar bank arrangements. Slopes of curves of Nusselt numbers versus Reynolds' numbers from Pierson's results are characteristically greater than those from Huge's data, and in addition there is a tendency to curvature or a change in slope of the former, as compared with Huge's linear characteristics.

Elimination of these differences, by identification of their causes if possible, is a necessary preliminary to correlation of the convection data. The causes presumably are associated with the differences in experimental methods used by the two investigators,[3,4] and these include factors of tube size and position, of air pressure, and of variations in tube temperature with rate of air flow.

All of these possibilities have been examined without disclosing explanation of the discrepancies in question. Effects of tube size or position probably would be reflected more markedly in flow-resistance results, where remarkably good agreement has been shown. No perceptible effect of changes in air pressure in Pierson's tests[4] are indicated by examination of the data on either convection or flow resistance.

Greatest uncertainty does, however, exist in the treatment of temperature effects. A great deal of study has been given to this phase of the problem, without reaching any satisfactory solution. The problem involves the different directions of the heat transfer in Pierson's and in Huge's small-size apparatus,[3,4] and the wide variations in tube temperatures with rate of air flow in Pierson's as compared with substantially constant tube temperature in Huge's apparatus. The good agreement of results of Huge's tests[3] on large-size banks with air heating with those on small-size banks with air cooling, indicate that the direction of heat transfer has no effect on its magnitude.

Variation of tube temperature in Pierson's apparatus[4] with rate of air flow appears to be the most probable cause of the qualitative discrepancies of results. This conclusion is not well-supported by the data from Pierson's tests[4] with reduced electrical input, which are in good agreement with normal input results in most cases, but the temperature ranges are not sufficient to arrive at any better conclusion. For the same reason, it has not been possible to derive any empiric treatment of temperature effect.

A number of analytical devices have been tried, including that recommended by Sieder (11). None of these result in any general improvement over the conventional procedure which has been adopted. Lacking any better treatment, all data in this report are evaluated by taking thermal conductivity k and absolute viscosity μ of Equation [2] at a temperature which is the average of tube and air-stream mean temperatures; i.e., at average boundary-layer or film temperature. This procedure brings the data from the several sources into better agreement quantitatively than the alternative simple procedure of evaluating physical properties at average fluid temperature, but it is interesting to note that a change to the latter basis would bring the slopes of the curves nearly into agreement.

It is obvious from the foregoing that the discrepancies in slopes of the curves of Nusselt numbers versus Reynolds' numbers are due to inadequate analysis, rather than to inaccuracies of the original experimental observations which are believed to be of high order. It is probable that a satisfactory solution will not be found without further experiment covering much greater ranges of temperature.

Modification to Pierson's Data

In the absence of a method of correlation which would bring the data into both qualitative and quantitative agreement, it is necessary to adopt as basic those results which most nearly represent conditions of application or use, and bring the others consistently into agreement with them by any available means. The field of application of chief interest in the present instance (steam boilers and related heat-transfer equipment) is more nearly represented by temperature conditions of Huge's experiments,[3] in which tube temperatures are substantially independent

of thermal load on the surface, and where thermal current varies with rate of flow. Huge's data[3] were therefore accepted as basic in respect to characteristic values of slope of the curves of Nusselt numbers versus Reynolds' numbers and a relation between slopes of these curves from the two sources was sought as a basis for consistent modification of Pierson's data.[4] It should be noted that relative values finally are of more significance than absolute values because of the usual necessity of adjustments, due to the combined effect of convection and other resistances, to bring them into line with field tests.

Study of the data from the two sources[3,4] reveals an interesting relation between slopes of curves of Nusselt numbers versus Reynolds' numbers, and slopes of straight-line approximate averages of curves of friction factors versus Reynolds' numbers (here, as in preceding discussions, slope refers to plots on logarithmic coordinates). Referring to Fig. 8, the data from Huge's tests[3] are plotted at A, and there is a definite trend as shown by the mean curve. Corresponding plots from Pierson's data[4] are shown at B, with a mean curve, and for comparison, the mean curve from Huge's data[3] is added. The parallelism between the two is noteworthy, as shown by a third curve drawn by deducting 0.04 from the mean curve of Pierson's data.[4]

The latter curve was used as the basis for modification of all of Pierson's Nusselt-Reynolds' number characteristics. Although this represents an average reduction of 0.04 in values of the exponent of Reynolds' number in Equation [2]; in individual cases the reduction varies from zero to 0.1. This device is a convenient generalization of trends rather than a rigid empiric relation. Refinement is not justified, as a rigid interpretation of Fig. 8 would require that the convection characteristics be nonlinear, corresponding to the friction-factor curves.

There is a definite trend toward quantative agreement of Pierson's and Huge's values of Nusselt numbers in the neighborhood of a Reynolds number of 30,000, and this was adopted as general. The modifications to Pierson's data[4] finally took the form of passing a straight line through the original Pierson curves at a Reynolds number of 30,000, the new relation of the curves of Nusselt numbers versus Reynolds' numbers being given slopes according to the curve marked "Basis of Modification," shown in Fig. 8.

This procedure rather arbitrarily disregards the sharp downward break of original curves of Nusselt numbers versus Reynolds' numbers at low rates of flow for a few arrangements. Here again there is considerable uncertainty associated with causes, and the neglect of this circumstance reflects only the author's judgment as to permissible license in modifying the data for application. Fortunately, only a few arrangements are affected, including several that are not very desirable from other considerations.

Final Correlation of Convection Data

For correlation of convection data in relation to tube spacing, and for working purposes as well, it is convenient to adopt as basic a well-established convection factor for some given arrangement and spacing, and convert the values for other arrangements to the form of "arrangement factors," defined as

$$\left.\begin{aligned} F_a &= \frac{U_c}{(U_c)\text{ Base}} \\ F_a &= \frac{Nu}{(Nu)\text{ Base}} \end{aligned}\right\} \qquad [7]$$

when mass flow, temperature, fluid, and diameter are constant, and where F_a = arrangement factor for convection heat transfer; U_c and Nu = convection conductance and Nusselt number for any given tube arrangement; and (U_c) Base and (Nu) Base are values for a reference arrangement, for which average results for the $1\frac{3}{4} \times 2$ in-line arrangement are here selected.

Values of the arrangement factor have been calculated from Huge's[3] and Pierson's[4] data, the latter modified as described previously, and plotted and cross-plotted in relation to spacing. Values from the two sources have been averaged where discrepancies exist, these being small in most cases. There is no apparent systematic difference in values from the two sources considered.

Final averaged values of arrangement factor F_a are shown in Fig. 9 for in-line arrangements, and in Figs. 10 and 11 for staggered arrangements, transverse-flow area being the minimum in the former and diagonal in the latter. In all cases, mass flow is calculated corresponding to the minimum flow area, whether this occurs in transverse or in diagonal openings.

TABLE 1 VALUES OF B AND (m) IN THE GENERAL RELATION $U_cD_t/k = B(GD_t/\mu)^m$

Transverse spacing = S_T/D_t	1.25		1.5		2		3	
Longitudinal spacing = S_L/D_t	B	m	B	m	B	m	B	m
Staggered								
0.600	...	...	...	...	...	...	0.213	0.636
0.900	...	...	...	...	0.446	0.571	0.401	0.581
1.000	...	...	0.497	0.558	...	...	....	...
1.125	...	...	...	...	0.478	0.565	0.518	0.560
1.125	0.518	0.556	0.505	0.554	0.519	0.556	0.522	0.562
1.500	0.451	0.568	0.460	0.562	0.452	0.568	0.488	0.568
2 000	0.404	0.572	0.416	0.568	0.482	0.556	0.449	0.570
3.000	0.310	0.592	0.356	0.580	0.440	0.562	0.421	0.574
In-line								
1.250	0.348	0.592	0.275	0.608	0.100	0.704	0.0633	0.752
1.500	0.367	0.586	0 250	0 620	0.101	0.702	0.0678	0.744
2.000	0.418	0.570	0.299	0.602	0.229	0.632	0.198	0.648
3.000	0.290	0.601	0.357	0.584	0.374	0.581	0.286	0.608

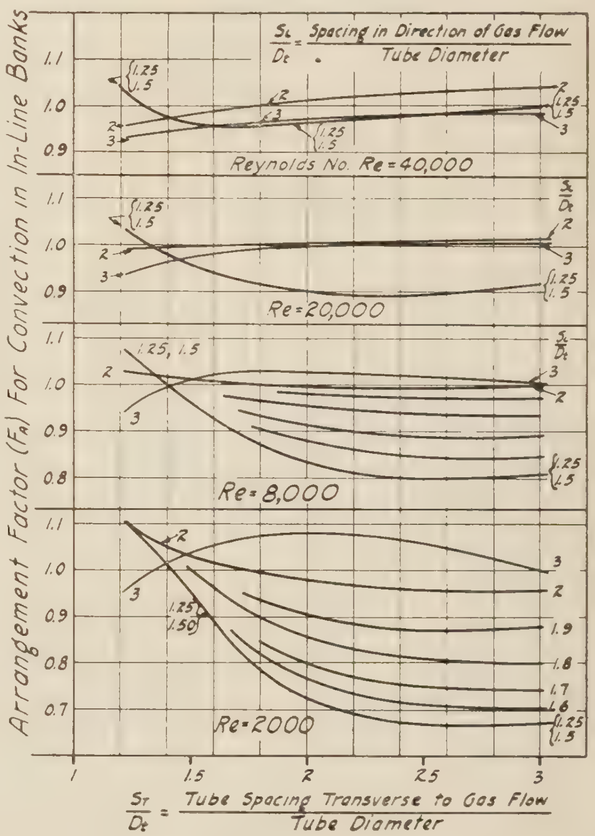

FIG. 9 ARRANGEMENT FACTOR F_a FOR CONVECTION HEAT TRANSFER FOR IN-LINE TUBE BANKS

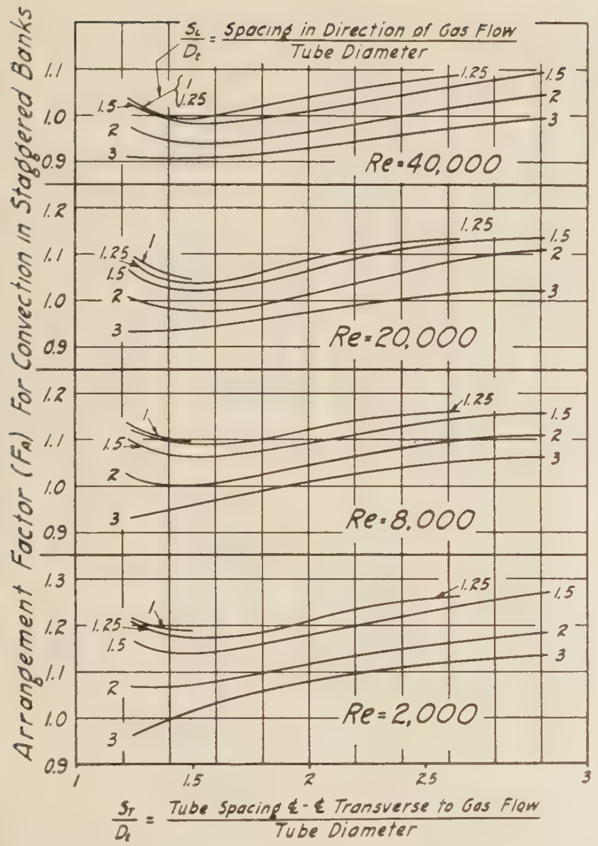

Fig. 10 Arrangement Factor F_a for Convection Heat Transfer for Staggered-Tube Banks With Minimum Gas-Flow Area in Transverse Openings

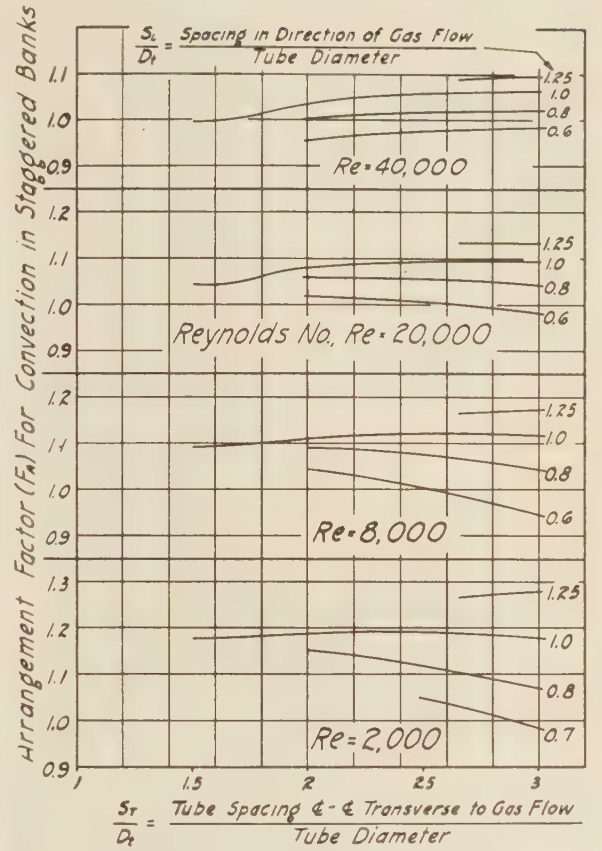

Fig. 11 Arrangement Factor F_a for Convection-Heat Transfer for Staggered-Tube Banks With Minimum Gas-Flow Area in Diagonal Openings

This makes a convenient working basis, the general relation being the product of the convection-conductance equation for the average curve of the $1^3/_4 \times 2$ in-line arrangements, and the arrangement factor.

$$U_c = 0.284F_a \left[\frac{k}{\mu^{0.61}}\right] \frac{G^{0.61}}{D_t^{0.39}} \qquad [8]$$

Arrangement factor F_a is taken from Figs. 8, 9, or 10. Thermal conductivity k and absolute viscosity μ are evaluated at the average of gas and tube mean temperatures.

The convection data are summarized in another form in Table 1, which gives values of the constant B and exponent m in the general relation given as Equation [2]. The data of Table 1 correspond to those given by the combination of Equation [8] and F_a values of Figs. 9, 10, and 11; i.e., they represent the final relations after modifications, fairing, and averaging as described previously.

Equation [8] applies strictly to banks ten rows in depth. For banks of lesser depth, convection conductance is reduced, and Pierson[4] has given correction factors in Fig. 17 of his paper[4] for specific in-line and staggered arrangements. In view of the disparity between values of the correction factors found by actual variation of depth of bank, and by calculations from measured conductances at individual rows of ten-deep banks, these curves may be applied to any arrangement for lack of better information. Probably the factors vary with spacings of tubes, but specific data are not now available.

The progressive increase in convection conductance from row to row and continuing to about the fifth row, where it appears that steady conditions are established, must result from generation of mass disturbances of the stream. These disturbances persist for some distance downstream of the bank, and tests by Reiher (12) and by Griffiths and Awbery (13) indicate no substantial dissipation of these disturbance effects five tube diameters downstream from a single row of tubes. Unless a substantial distance separates the first row of a bank from the last row of another bank preceding it, it is advisable to make no correction for depth of the second bank.

Comparisons With Other Data

Comparisons have been made in Fig. 12 of the convection data, from Equation [8] and Figs. 9, 10, and 11 suitably adjusted for depth of bank, with data reported by other investigators, taken from literature sources (12, 13, 14, 15, 16, 17, 18, and 19). Space is not available in this paper for review of the methods of determination of these data, but this material is available in the original sources and with exceptions, in some of the texts. Lindmark's report (14) has been translated from the Swedish, and is not available in the English literature. Griffiths and Awbery (13) have published their data more recently than any of the available texts written in English. The data reported by Thoma (15) from mass-transfer tests have been omitted, as they are superseded by those of Lohrisch (16) who improved the technique and used larger apparatus.

All of these data have been recalculated to the same basis, as far as possible. This has involved a great deal of labor and some

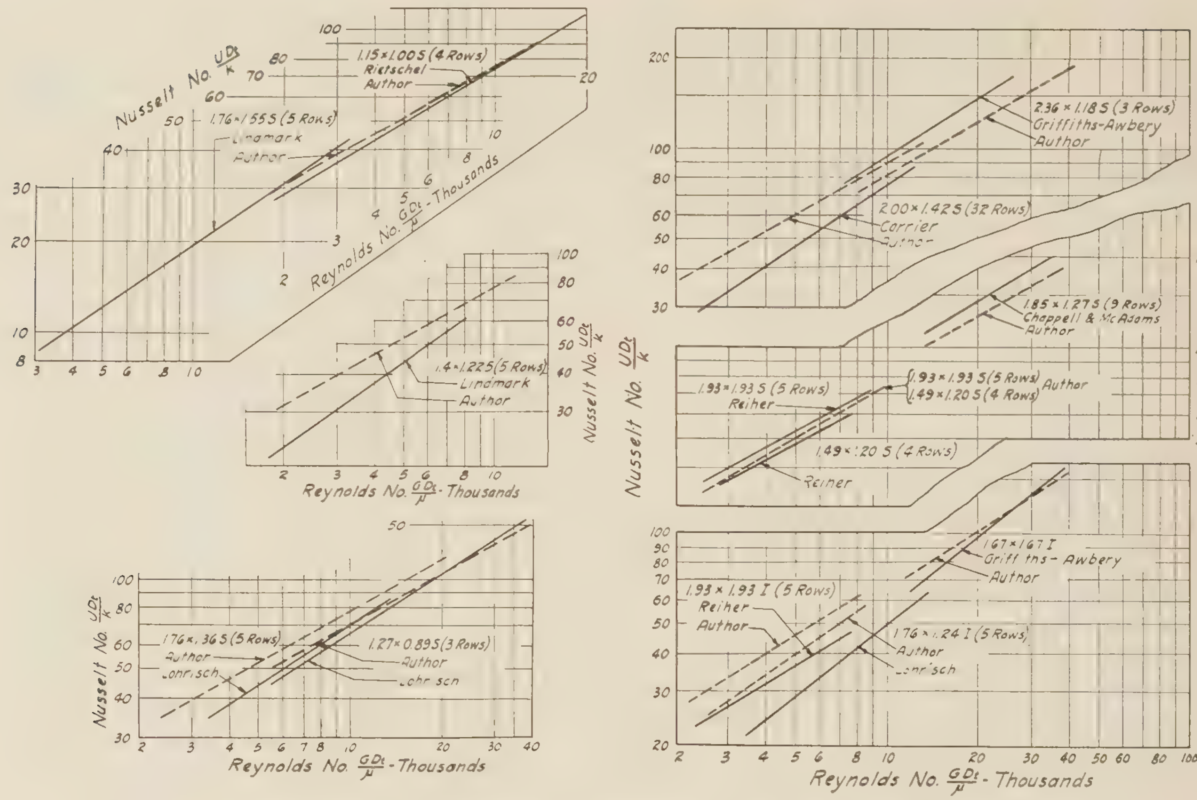

Fig. 12 Comparisons of Convection-Heat Transfer Data With Those of Other Investigators

assumptions, since the original test values are not always given. Attention is called to Reiher's results (12) in particular, which have been widely accepted and form the basis of recommendations in some texts. Reiher's very unorthodox definition of the temperature difference between gases and tubes apparently has been overlooked by most authorities, and has made it necessary to recalculate all of his results.

Not much of value can be gleaned from these comparisons, since no consistent trends are indicated. No interchecks between literature data are possible because of differing arrangements, although errors must occur in some of them. The comparisons with Lindmark's values for 1.40 × 1.22 banks, and with Reiher's for 1.49 × 1.20 banks, both staggered, may be noted. Agreement with the first is poor, and with the second excellent, and since there should be no great difference between values for these two arrangements, the question arises as to which should be accepted.

4 GENERAL CONCLUSIONS

The most important over-all result of this new work is the large amount of reliable information on relative performances of tube banks of a wide variety of arrangements. The resulting greater certainty of design calculations should make for better balance and economy of design, and for lower service costs to meet guarantees if the data are properly applied.

Within certain limits the value of model testing is established. For this type of equipment, extreme accuracy in establishing all dimensions is necessary if size is reduced below one-quarter scale, to avoid magnified effects of errors. As far as practicable, the conditions of experimental measurements should simulate those of expected application. The latter conclusion is not new, but is emphasized by the difficulties which have been encountered in preparing these data for application, as discussed previously.

Considerable attention has been given by some investigators, notably Colburn in this country and Prandtl in Germany, to mathematical relations between convection-heat-transfer rate and flow resistance. No attempt has been made in the present study, to explore this possibility but it appears on inspection that the two are not related in the sense that the first is a function of the second. For present purposes the completeness of the data on both convection and flow resistance vitiates to a great extent the practical value of such a relation.

Applications to Design Calculations

As already indicated, the data on flow resistance are applicable directly to calculation of pressure drop in commercial cross-flow tube banks, with proper provisions where necessary for unequal flow across banks in which "lanes" or other irregularities occur. Theoretically, they may also be applied to turbulent flow of liquids, and this is supported by the comparison of Figs. 6 and 7 with Sieder's (8) data for several liquids.

The utility of the convection data is of highest order when they are incorporated in a complete analytical treatment of thermal performance; one which identifies and properly combines the effects of major thermal resistances. These are in general gaseous radiation in intertube spaces in parallel with convection, the corresponding gas-boundary resistance being in series with resistances of dirt, soot, scale, coke, or other solid deposits on tube surfaces and with that at the boundary of the receiver fluid. When other resistances are relatively large, the accuracy of determination of the over-all resistance or its inverse conductance, is subject to the order of accuracy of predictions of individual resistance values.

The utility of these data for original design may be enhanced

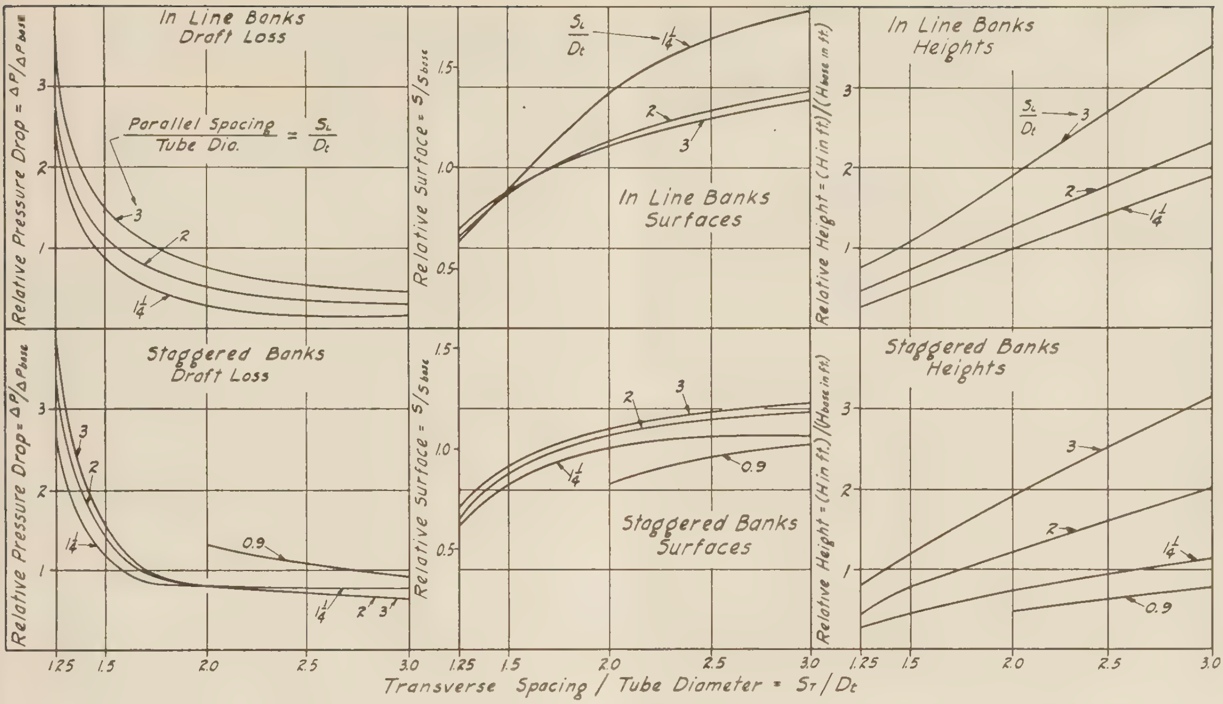

FIG. 13 RELATIVE DRAFT LOSSES AND DIMENSIONS OF CROSS-FLOW TUBE BANKS FOR FIXED THERMAL PERFORMANCE AND FIXED PLAN DIMENSIONS

by combining the flow-resistance and heat-transfer effects to determine optimum spacing for design conditions which are ordinarily imposed in a given class of work. For instance, it is frequently necessary to select a bank of tubes to occupy a given plan area, set by building columns or connections to related equipment, and to absorb a specified quantity of heat at a specified load or heat input to the bank in the gases. Assuming the heat transfer to be by convection only, or substantially so, tube spacings affect draft loss, required heating-surface areas, and heights as shown in Fig. 13, for both staggered and in-line arrangements. The effects of variation in mass flow with changes in tube spacing, under the stated condition of constant weight of gas per hour, are of course included.

Substantial savings in heating surface are indicated at close transverse spacings, but are accompanied by high draft losses. Some rough estimate of relative costs of fans and fan power against additional heating surface, casing, supports, and space is necessary, but a reasonably good first selection of tube spacing probably can be made by inspection. If detailed calculations are made for the selected tube arrangement, the effects of variation from it may be determined by ratios from the curves of Fig. 13.

The foregoing is of course a rudimentary analysis of a phase of the general design problem, but is offered in illustration of a direction of extension of the data toward broader utility. The ultimate solution would of course include many other factors of costs, both initial and operating. It is desirable to reiterate that it has been assumed that convection controls the heat transfer. Occurrence of other relatively large resistances in parallel or in series will lead to some modifications of conclusions.

RELATED PROBLEMS OF APPLICATION

The value of these new data will be greatest to users who are equipped to apply them in a full analytical treatment of the problem. In this connection, it is desirable to summarize the relation of these data to the other phases of the problem.

The thermal performance of cross-flow tube banks depends on the over-all value of the thermal resistance to transfer of heat from a source fluid on one side of the tube wall to a receiver fluid on the other side. The over-all resistance is the sum of three partial resistances in series, which are of different kinds; that is, the resistances (*a*) at the boundary of the fluid sweeping the external surfaces of the tubes, (*b*) of the tube wall and any solid deposits which may accumulate on both of its faces in operation, and (*c*) at the boundary of the fluid inside the tubes flowing parallel to the tube walls, and which may be heating or cooling, boiling or condensing.

The resistance at the boundary of the external cross-sweeping fluid, assuming the latter to be a gas, is the reciprocal of a thermal conductance, which is the sum of a convection part and a radiation part. For air, convection conductance may be taken directly from the data which have been given in this paper, and for other common gases may be estimated with reasonable accuracy from these data by application of Equation [1], if physical constants are known. For liquids in turbulent flow, other basic data must be combined with the arrangement factors of Figs. 9, 10, and 11.

In spite of the small values of effective gas-layer thicknesses in most tube banks, radiation between tubes and gases in intertube spaces is often a factor of importance in tube-bank performance. As a basis for calculations, there are available recent experimental measurements of radiation of carbon dioxide and steam by Hottel and Mangelsdorf (20), which have been confirmed approximately by Fishenden (21), but for water vapor are in marked disagreement with previous data by Schmidt (22). The Hottel and Mangelsdorf data may be accepted for present purposes, and values of radiant conductance may be calculated from the original values and suitably plotted for use. More pertinent to the

present problem is the lack of suitable simple methods of determining, for any given bank, the effective radiating mean thickness of the gas body between tubes. Hottel (23) has given values for certain defined conditions, but extension of these data is greatly to be desired. For the present, grossly approximate estimates are commonly used.

For the inside boundary conductance, or its inverse resistance, data are available as a basis for a general calculation procedure. These data are not of satisfactory accuracy for all circumstances of application, but it is generally true that an approximate value can always be determined. In many cases, the inside boundary resistance is so small relative to the other resistances in series, that it may be neglected entirely.

Few data of a fundamental nature are available on the thermal resistances of solid deposits normally occurring on tube surfaces. However, as the nature and extent of the fouling depends to a great extent on the fuel in direct-fired equipment and on operating practices, the most useful data on this resistance are those showing its net effect on performance, as determined from field tests. This is true of the present situation, but development of knowledge will lead to more detailed analysis of this phase of the problem.

The effective value of the fouling resistance (including tube-metal resistance when this is relatively small) is calculated from test data in any given case as follows: From known values of effective heating surface, and test data on heat transferred and mean temperature difference between source and receiver fluids, the over-all thermal conductance (heat-transfer rate) may be calculated, and its inverse is the mean over-all resistance. External and internal fluid-boundary resistances are calculated from fundamental data, mainly from literature sources, suitably analyzed and modified if necessary, and assumed to be correct in the absence of special information. The "wall resistance" is the difference between the test value of over-all resistance and the sum of the calculated values of resistances at the fluid boundary.

The greatest possible quantity of data should be collected on wall-resistance values, and related as far as possible to factors of influence such as fuel type, cleaning devices, and frequency of their operation, and for unfired apparatus to the type and stability of such factors as the fluids and velocities. When these data are combined with those previously established for fluid-boundary resistances, they form the basis of a sound and flexible calculation procedure, subject to broad application. Experience has shown that a general basic procedure of this sort is readily adaptable to routine calculation work, although superficially it may appear very cumbersome to engineers used to simple empiric relations. The empiric procedure is preferable for application to fixed-type units with limited ranges of proportions and operating conditions when new equipment substantially reproduces constructions for which test data are available, but is not safe otherwise.

One of the greatest virtues of a properly developed calculation procedure of the kind outlined, results from the transfer of necessary estimating or guess work from over-all to detailed values. The elements of judgment which must be introduced are more reliable under such a procedure by reason of the systematic correlation of past experience which is inherent in a sound analysis. It is greatly to be desired that refinement of the several phases of the calculation procedure be continued by further experiment and by improvement in accuracy of field testing.

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Industrial Instruments, Their Theory and Application

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This paper is complemental to the paper: "Automatic Regulators, Their Theory and Application," published previously in the A.S.M.E. Transactions (1).[3] The treatment of the material in this present paper parallels that given in the previous paper (1) for the reason that instruments and regulators have so much in common, there being both self-operated and servo-operated types in both cases. The principal difference between them is that the regulator reacts upon the variable while the instrument in general does not do so in such a way as to cause the possibility of hunting. Most of the material in this present paper is perforce old; however, the invasion of the industrial-instrument field by devices operated from a separate source of power has been so gradual that this modern situation should be emphasized at this time. This paper is a contribution toward that end.

IN THE process industries, industrial instruments, including regulators, are useful mainly as they contribute to the control of processes whether for the purpose of replacing operating personnel, for increasing process efficiency, or for both at once. Worthwhile increases of production with improved quality of product are common with modern instruments so used, many processes being entirely impracticable without the aid of the most highly improved measuring and controlling equipment.

The increasingly rigorous requirements of high continuous, as opposed to limited batch, production in the process industries have led to the gradual replacement of self-operated instruments by servo-operated instruments of higher accuracy, often coupled with more rapid indicating ability than is possible, without a separate source of power, in any instrument of reasonably small dimensions. The order of accuracy is commonly improved in such cases from within about 0.5 to 1 per cent of full scale with self-operated instruments to within about 0.05 to 0.1 per cent of full scale with servo-operated instruments of the null-method type. In other cases, servo-operated instruments of the "booster" type are widely used, possibly without any gain in accuracy, but for some other purpose as controlling, telemetering, or integrating, most of such instruments being of the cyclically-actuated type.

The excellence and low cost of the many small electric motors available today have led to the general adoption, for servo-operated instruments, of the synchronous self-starting motors and of the induction motors having an adjustable range of operating speeds. The latter type is especially suited for governing by the sensitive relays used in highly sensitive instruments.

A work such as this is complete only with the inclusion of the theory of rapid indication. This has been taken largely from a book by Drysdale and Jolley (2) which deals with the general conditions for securing promptitude with self-operated instruments. This theory is not strictly applicable to servo-operated instruments, but is still useful even though these may not follow the laws of simple harmonic motion, the use of series expressions often bringing the analysis within its teachings.

As with regulators, the most useful classification of instruments varies with the use to be made of it. The classification presented herein is incomplete but may serve to indicate the nature of the problem involved in the setting up of such classifications.

As to the scope of instrument classes to be considered by the A.S.M.E. Subcommittee on Industrial Instruments and Regulators, it seems desirable to exclude plug gages, micrometers and other such tools as well as devices for testing materials which should more properly be within the field of the American Society for Testing Materials, and also to exclude instruments for measuring electrical quantities falling within the domain of the American Institute of Electrical Engineers. However, it should be mutually desirable for the A.S.M.E., A.S.T.M., and A.I.E.E. to collaborate on the mechanical engineering problems involved. It seems proper that temperature responsive devices should be a concern of this Society even when they are of an electric servo-operated type, especially in view of the importance of such devices to the Process Industries Division of the A.S.M.E., and of the mechanical-performance characteristics of such instruments.

Scope and Definitions

The scope of this paper is limited to those classes of industrial instruments which are responsive to variable magnitudes and useful in connection with processing material, and in particular to their performance characteristics. Such typical variables are flow rate, level, pressure, temperature, and speed. Such instruments also include the measuring devices common to regulators even though the latter have no separate instrument for indicating, recording, or integrating. This paper is complemental to that on "Automatic Regulators" (1) and contains associated material on the "primary" elements sensitive to the variable and on the "secondary" elements in the instruments responsive to the impulses produced by such primary elements.

The following few definitions are presented to facilitate the reading of this paper rather than for offhand adoption, no attempt being made to cover comprehensively all terms used:

[1] Mechanical Engineer in Charge of Patents, C. J. Tagliabue Manufacturing Company, Mem. A.S.M.E. Mr. Smith attended the University of California from which he received the degrees of B.S. in 1919 and M.E. in 1932. After graduation he was employed by the Standard Oil Company of California on the testing and design of refining equipment. In 1922 he was engaged in private research at the University of California, and in 1923 entered the employ of the Builders Iron Foundry where he served as hydraulic engineer from 1926 to 1936. Since the latter date he has held his present position.

[2] Director of Research, C. J. Tagliabue Manufacturing Company. Mr. Fairchild was graduated from Rose Polytechnic Institute with a B.S. degree in 1912, and received the M.S. degree from the same Institute in 1920. From 1912 to 1914 he was employed as a chemical engineer with the Western Chemical Company, and from 1915 to 1926 was physicist at the National Bureau of Standards and chief of pyrometry section. Since 1926 he has held his present position.

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Process Industries Division and presented at the Meeting of The American Society of Mechanical Engineers, held at Erie, Pa., October 4 to 6, 1937.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1937, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

1 An industrial instrument is a device which has a portion that is either positioned or stressed in correspondence with a variable, such portion being used either to indicate or record some function of this variable or to control this or another variable.

2 A self-operated instrument is one actuated by the variable, either directly (for example, a Bourdon pressure gage) or indirectly by a dependent variable (for example, the vapor pressure in a fluid-containing closed-tube system responsive to temperature), but in either case without the use of power from another source or even from the same source used as though from a separate source.

3 A servo-operated instrument is one actuated by power from a separate source which is governed by a relay operated either by the variable itself or by a dependent variable.

4 Sensitivity is the ratio of the magnitude of the response of the indicating portion to a change of the variable.

5 The throttling band, or range, in a regulator comprising a controller governed by an instrument is inversely proportional to the sensitivity of the whole device, provided the sensitivity be related to the throttling band as, for example, with a variable radius adjustment, without discontinuity, between the nozzle and the flapper fulcrum in an air-operated controller; the throttling range being the range of values of the variable (either actual or indicated, as specified) necessary to send the controller to its limits.

6 Error is the difference between the value of the variable and its indication.

7 The dead zone is the extreme range of values of the variable possible without moving the indicator from a given position, in the absence of abnormal vibration; the dead zone is ordinarily twice the frictional error.

General Theory

Instruments for indicating and regulators for controlling variables have so much in common that undue emphasis is generally laid upon their principal difference: In an instrument, a member is positioned in correspondence with a variable without affecting the latter, while in a regulator this member acts upon the measured variable, or another, either directly or indirectly. In each case, since a member is positioned directly by the variable or by power from a separate source, instruments, like regulators, may be classified as self-operated and servo-operated, the latter also being known as the relay type. The self-operated instrument responds to a change of the variable in a manner depending upon its sensitivity, actuating force, inertia, damping, and mechanical friction, but always stably, that is, the motion is not self-perpetuating, although that of a self-operating regulator may be. On the other hand, servo-operated instruments are like servo-operated regulators in that they may be either stable or unstable; with an unstable instrument, the hunting frequency depends primarily upon the characteristics of the instrument itself, while with an unstable regulator the frequency depends more upon the characteristics of the system to be controlled.

Due to the inertia of its motor, the characteristics of a servo-operated instrument differ from those of a self-operated instrument since the former is generally driven at nearly constant speed up to within close limits of the proper point, while the latter may be actuated less strongly as the normal position is approached, a characteristic of a harmonic relation. In other words, a chief problem of the servo-operated instrument is to slow down the indicating member as it approaches the proper value so as to minimize the effects of inertia. While various electrical and mechanical solutions of this problem exist, still it may be noteworthy that the use of light electro-magnetic clutches goes far in reducing the difficulty where space and economic considerations justify this complication. Again, this problem is less serious in self-operating instruments for another reason: An equally high order of accuracy coupled with quick action is seldom required. In such instruments, however, if the same accuracy and speed were required, the same problem would be so serious as to make an economic solution generally impracticable; at any rate, the special damping means required would complicate such instruments to the point that they would no longer be simple instruments even though they were still self-operating.

Static Relations

The dead zone may be determined, assuming a sufficiently sensitive standard to be available for comparison, by bringing the variable gradually to the same value first from one, and then from the opposite, direction in an effort to approximate statically the relation between the biasing, or restoring, force and the friction. However, the average should not be used as the calibration, due to the difference between starting and running friction and also to the fact that an imperceptible vibration may cause one of the limits of the deadzone so determined to be in error. Instead, it is generally accepted that more reliable values of the calibration are obtained by lightly jarring the instrument while approaching the same value of the variable from both directions. In testing, the values indicated by the instrument are generally used instead of those of the standard, that is, the instrument being calibrated is brought to the same mark from both directions while the standard is read. However, the practice depends somewhat on the type of standard available; for example, obviously if only a few points were available, such as freezing and boiling points, the instrument reading must be used, while with a flow meter calibrated against a manometer having an engine-divided scale with closely spaced graduations, the flow meter may be more closely checked exactly on its own fewer graduations, instead of introducing unnecessary errors in interpolation. Care must be exercised where asymmetry of the operating mechanism exists to avoid inertia effects, as where a roller may be jarred away from its cam or along it in one direction, but not into it or along it in the other. However, in addition to the calibration obtained in the presence of such jarring, which calibration does not tell the whole story, it is also necessary to check the width of the dead zone to determine the possible accuracy of the instrument in service. Numerous precision instruments are fitted with small electrical vibrators to decrease the dead zone.

Accuracy is frequently an item in the specifications for an instrument at the time of its purchase, this still being often stated as "within a guaranteed error for any single reading." This standard may be adequate for instruments used in determining distances, weights, or other fixed quantities. However, for industrial instruments used in the process industries, which must measure varying quantities (variable with time), it is more useful to specify the accuracy as equally liable to exceed or fall below the limit of error known as the mean probable error. One advantage of this method is the fact that such a value has a sound method of calculation behind it; further, by the use of a comparatively simple tabulation, the guaranteed value may be reliably checked. The first-mentioned method is unfair to the more conservative manufacturers of industrial instruments, who give unreasonably large errors as possible with their instruments rather than risk being placed in the position of having to lower a guarantee which has once been published. In other words, it is much more difficult to determine the extreme value of error that an instrument will never exceed than to determine the error to be expected, taking readings day after day, which is the way that most industrial instruments are used. In the case of integrators, this most nearly corresponds with the cumulative error "wedge."

Where a uniformly graduated scale is used, it is logical to state

the accuracy in percentage of the entire scale. However, when a logarithmic scale is used, or a flow-discharge (or other quantity-rate) scale, where a low rate can be maintained for extended periods, the guarantee may be stated in percentage of the reading of the variable. In the latter case, the user should understand the engineering problem faced at low rates, especially with flow meters where the discharge follows the square root of the head, so that an increasing multiplication must be used as the head decreases. Thus at one tenth of the maximum rate, the head is only 1 per cent of the maximum head and where plus or minus 2 per cent is there required, this corresponds with an error of only 0.04 per cent of the maximum head. Such a meter obviously must be accurately adjusted at the minimum operating rate, and further be maintained in excellent condition. The foregoing computation is particularly useful in comparing the tasks of electrical meters and flow meters and their necessary relative excellence.

While a flow meter's indicator or recorder is usually uniformly graduated to facilitate integration, yet, since they indicate or record a rate, their accuracy is customarily stated in terms of the reading of the variable. The limitations of this method may be brought out by considering a chart having a 4-in. band. At the 10 per cent rate, the pen is only 0.4 in. from zero, 2 per cent of which is 0.008 in. This limit is quite beyond that which the average eye can see and also beyond the accuracy of the reference lines on the chart, considering printing and engraving tolerances and the effects of variations of atmospheric humidity on the paper; thus, a guarantee of a range of twenty to one to within plus or minus 2 per cent with such a chart would require that the line be read to within 0.004 in. Hence, it seems evident that the integrator alone should be required to meet such a guarantee, as a mere matter of common sense. Even for the integrator to do so, assuming the time mechanism to be perfectly accurate and the integrator perfectly set, requires an accurate response to a differential of only 0.23 per cent of the maximum head, so that for a maximum head of 20 in. Hg, the instrument must follow a change of only 0.002 in. in the differential of mercury or 0.0252 in. of water, which would be caused by a change of 10 F in a column of water only 1.5 ft high.

Accuracy is expressed customarily by the inaccuracy or error; thus, "within 2 per cent" is universally used instead of the possibly more logical but less forceful 98 per cent. The 2 per cent means plus or minus 2 per cent, giving a zone of error of 4 per cent. Although this method is not exact, it is precise enough for practical purposes as long as the errors are of the small order common with industrial instruments, which are generally likely to be more accurate and reliable than home-made test instruments improvised for a single test.

The indicated frictional error, one half of the dead zone, is equal to the frictional force s' divided by the force per unit of the scale, or

$$e_i = \frac{s'}{dF/dy} = s'\frac{dy}{dF} \qquad [1]$$

or, expressed in terms of the variable x, where the sensitivity

$$b' = dy/dx \qquad [2]$$

the error is

$$e_i = \frac{s'b'}{dF\,dx} \qquad [3]$$

Thus, a self-operating meter is functionally at a disadvantage as compared with one powerfully actuated by auxiliary energy, although the former may have the practical advantage of greater simplicity and lower operating maintenance. Also, the added motor must be provided with means to keep it from over-running the proper point and from hunting. Null-method meters are increasingly available for industrial purposes, largely due to the increased facility in electrical technique and the reliability of this source of power and its associated equipment. Further, another class of meters having power-driven cyclically operated recorders, telemeters, or impulse-actuated controllers, unloads the indicating member during a portion of each cycle so that a relatively small force will serve to position it; with these, the cyclically movable device may provide a method of obtaining a desired non-lineal relation between the variable and its indication; for example, where a square root or other relation requiring high multiplication is involved.

Where any of these meters are used to govern the control means of a regulator, the sensitivity s_r of the whole regulator may be expressed as the product of the sensitivities of the meter and controller s_m and s_c, respectively, or

$$s_r = s_m\, s_c \qquad [3a]$$

With a hunting regulator, the meter may be in another phase than that of the controller so that their deflections should be added vectorially rather than algebraically, this point coming up mainly in temperature regulation where the magnitude of the several lags may be great and of widely differing orders. A metering lag which augments the effect of process lag tends to produce hunting while a governing lag, i.e., between the meter and controller, tends to produce stable control by at least partially compensating for process lag. Somewhat similarly, the higher the speed of a servo-operated meter (without hunting of the meter itself) and the lower the controller speed, the less is the tendency for the whole regulator to hunt. While offhand there may not appear to be a close relation between delaying the action of a controller and slowing it down, actually this point must be considered in analyzing regulators. These remarks have been included to show the intimate relations existing between the performance characteristics of a meter and of the controller which it governs.

Dynamic Relations

Before expressing dynamic relations mathematically, it may be mentioned that metal springs and bellows exhibit appreciable hysteresis; a similar drifting effect occurring in flow meters where mercury cannot flow readily from one well to the other for some reason. Thus, a U-tube balance with small connecting pipes will have a greater apparent accuracy than ultimate accuracy. Its apparent liveliness is due to the fact that the entire U-tube swings. This is a spectacular exception to most flow-measuring instruments, which are more likely to be over-damped than under-damped.

It is preferable to have a meter slightly under-damped rather than over-damped. However, as previously emphasized, a complex meter may be apparently under-damped in regard to momentary swings and still be highly over-damped ultimately, an undesirable condition on the whole since the user is led to rely on the momentary fictitious value as the actual, which it is not.

Besides molecular damping, either solid or fluid, and electro-magnetic damping, there is also friction between adjacent solids. Solid friction does not vary much with velocity (although starting friction exceeds stopping friction) and widens the dead zone, while fluid and electro-magnetic damping increase with some function of velocity but do not widen the ultimate dead zone except as they affect the rate at which the indicator swings into this zone and thus affect the average, or probable, mean error. This mean error has more pertinence than the maximum possible error in most industrial applications and has gradually come to be what is meant in user's accuracy specifications; the probable error being today more popular for test purposes than the for-

merly popular maximum possible error. However, it may be remarked that the S.A.M.A. still officially clings to the maximum possible limit of error as a basis for guarantees.

The ideal instrument should indicate changes of the variable without lag or oscillation.[4] Due to inertia, any actual instrument will lag behind the variations, and oscillate excessively in the absence of suitable damping. Translational and rotational systems follow the same laws, the mass m in grams and the acceleration d^2y/dt^2 in centimeters per second per second of lineal systems corresponding to the moment of inertia I (g-cm²) and angular acceleration $d^2\gamma/dt^2$ in radians per second per second, respectively. In instrument work, forces are usually so small that the use of centimeter-gram-second units is a real convenience. The authors gratefully acknowledge the assistance of their associate, V. L. Parsegian, in the preparation of this portion.

The nomenclature used in this paper is as follows:

Translational (for text):

- Y = linear deflection, cm
- Y_0 = initial deflection, cm
- Y_e = one half of dead zone, cm
- m = mass (linear inertia), g
- T = period of a complete oscillation, sec
- b = restoring (or "biasing") force per unit displacement, dynes per cm
- N = damping force for a unit velocity, dynes sec per cm
- N_c = damping force for a unit velocity for critical damping, dynes sec per cm
- F_e = frictional force (static assumed equal to dynamic), dynes

Rotational (for Figs. 1 to 11, inclusive):

- γ = angular deflection, radians
- γ_0 = initial deflection, radians
- γ_e = F_k/B = one half of dead zone, radians
- I = moment of inertia, g cm²
- T_r = period of complete oscillation, sec
- B = restoring (or "biasing") torque per unit deflection, dynes cm sec per radian
- N_r = damping torque per unit angular velocity, dynes cm sec per cm
- N_{rc} = damping torque per unit angular velocity for critical damping, dynes cm sec per radian
- F_k = frictional torque (static assumed equal to dynamic), dyne cm

Common to translational and rotational:

- M = ratio of damped to undamped period
- λ = logarithmic decrement

1 *Inertia, No Friction, No Damping.* Consider a system having inertia but no friction or damping force. Under a force F the movement is given by Newton's second law as

$$F = m(d^2y/dt^2) \qquad [4]$$

and if this movement were opposed by a spring or similar device the restoring force would be

$$F = by \qquad [5]$$

the two opposing each other in the form

$$m(d^2y/dt^2) - by = 0 \qquad [6]$$

It is easily shown that Equation [6] expresses a system having simple harmonic motion, for if

$$y = \cos \omega t \qquad [7]$$

expresses a harmonic movement of a point oscillating along the y axis, having a velocity

$$(dy/dt) = -\omega \sin \omega t \qquad [8]$$

and having an acceleration

$$(d^2y/dt^2) = -\omega^2 \cos \omega t \qquad [9]$$

then, by combining Equations [7] and [9], we obtain

$$d^2y/dt^2 + \omega^2 y = 0 \qquad [10]$$

where, comparing with Equation [6]

$$\omega = \sqrt{(b/m)} \qquad [11]$$

and the period of each complete cycle is given by

$$T = 2\pi/\omega = 2\pi\sqrt{(m/b)} \qquad [12]$$

Upon an instantaneous change of the variable, the indicator travels twice the change, and thereafter oscillates between this higher value and the initial value of y.

With a rotational system, having a torsion constant B in dynes $\times$ centimeters/radian, the torque in dyne-centimeters is

$$T_r = 2\pi\sqrt{(I/B)} \qquad [13]$$

Fig. 1 illustrates the relation between the natural period T_r and the inertia I for the various values of the restoring constant B. This figure applies equally well to a translational system if mass m in grams were substituted for the inertia I in the abscissa, and an opposing force b in dynes were substituted for each of the values of B; for example, for an indicating system having m = 6 g, b = 20 dynes per cm, the period according to the Fig. 1 is about 3.45 sec. Checking this value with Equation [12] we find that $T_r = 2\pi\sqrt{(6/20)}$ = 3.43 sec.

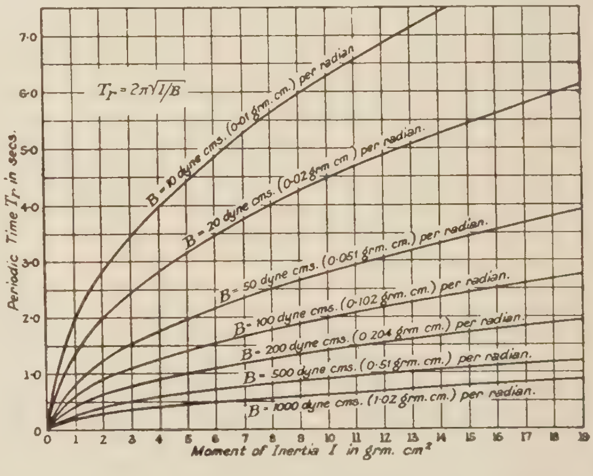

Fig. 1 Relation Between Periodic Time T_r and Moment of Inertia I for Oscillating Systems

2 *Friction, No Inertia, No Damping.* The friction in this case is constant, always opposing motion and thereby determining a dead zone $2y_e$ wide as determined from Hooke's Law wherein

$$y_e = F_e/b \qquad [15]$$

Upon being disturbed and released, the indicator would return instantly to the edge of this dead zone.

3 *Damping, No Inertia, No Friction.* In this case, the damping

[4] Figs. 1 to 9, inclusive, and much of their context were taken from the book "Electrical Measuring Instruments," by C. V. Drysdale and A. C. Jolley, Ernest Benn, Ltd., London, 1924, pp. 75–108, and are reproduced in this paper through the courtesy of the authors and publisher.

force of N times the velocity (fluid friction) opposes the restoring force, or

$$dy/dt = -(b/N)\, y \quad \ldots\ldots\ldots\ldots\ldots [16]$$

the solution of which is

$$y = y_0 e^{-(b/N)t} = y_0 e^{-\alpha t} \quad \ldots\ldots\ldots\ldots\ldots [17]$$

where

$$\alpha = b/N \quad \ldots\ldots\ldots\ldots\ldots [18]$$

From Equation [16] it is seen that the velocity decreases to zero as the indicator approaches the normal position (where $y = 0$); consequently, there is no overrun. Fig. 2 illustrates the approach to exact balance for a rotational system having such characteristics. Similarly, it may hold true for a translational system having a biasing constant $b = 200$ dynes per cm and a damping constant $N = 100$ dynes per cm/sec if the deflection be measured as y cm.

Fig. 17 illustrates the behavior of the solution [17] when plotted on semi-log paper. Although the figure is applied here to a temperature recorder it may be more easily adapted, because of the semi-logarithmic plot, to practical problems than can Fig. 2.

Useful equations may be derived from Equations [16], [17], and [18] as follows

$$\log_e \frac{y}{y_0} = \frac{b}{N}\, t = \frac{1}{0.4343} \log_{10} \frac{y}{y_0} \quad \ldots\ldots [18a]$$

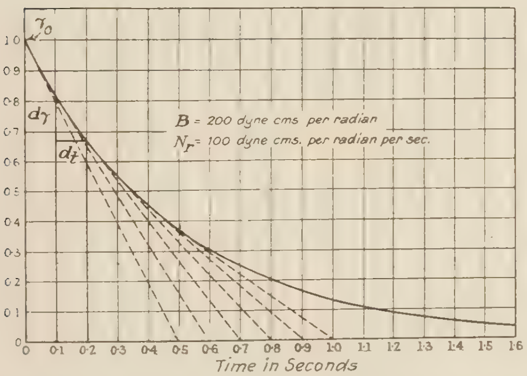

FIG. 2 DECREASE IN DEFLECTION FOR DAMPED SYSTEM WITHOUT INERTIA—SEE FIG. 17

or

$$\log_{10} \frac{y}{y_0} = -0.4343\, \frac{b}{N}\, t \quad \ldots\ldots\ldots\ldots [18b]$$

4 Damping and Friction, No Inertia. In this case, the friction determines a dead zone y_e, and both the damping and friction tend to oppose the restoring force or

$$\frac{dy}{dt} = -\frac{b}{N}\,(y - y_e) \quad \ldots\ldots\ldots\ldots\ldots [19]$$

the solution of which is

$$y = (y_0 - y_e)e^{-(b/N)t} + y_e \quad \ldots\ldots\ldots\ldots [20]$$

Fig. 3 illustrates the movement to the edge of the dead zone for a rotating system. The same graph may be applied to the movement of a translational system having $b = 200$ dynes per cm, $N = 100$ dynes per cm/sec, and $F_K = 8$ dynes, if the deflection be measured as y cm.

5 Inertia and Friction, No Damping. As shown in Fig. 4, a deflected indicator oscillates, but the amplitude of oscillation is reduced by the width of the dead zone with each half-cycle swing. The energy of the system is ultimately absorbed sufficiently by the frictional losses, and the indicator comes to rest within the dead zone. Fig. 4 is again drawn for a rotational system but may be used for a translational system having $T = 1$ sec and $y_e = F_e/b = 0.04$ if the deflection be measured as y cm and the initial deflection is $y_0 = 1$ cm.

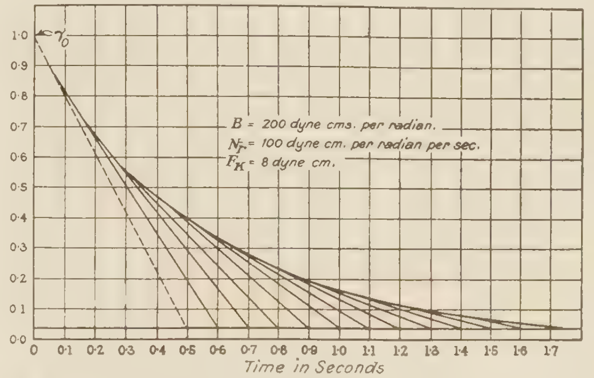

FIG. 3 DECREASE IN DEFLECTION FOR DAMPED SYSTEM WITH FRICTION BUT NO INERTIA

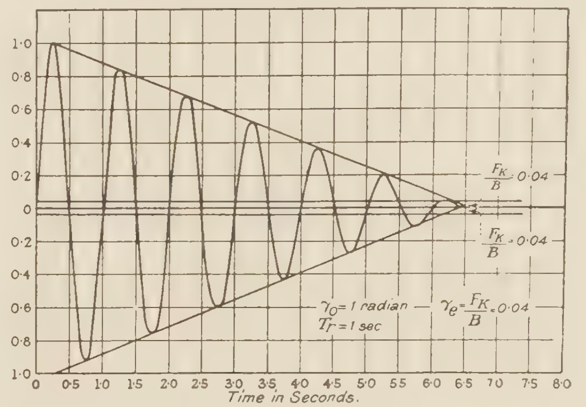

FIG. 4 OSCILLATING SYSTEM WITH INERTIA AND FRICTION

6 Inertia and Damping, No Friction. Equating the inertia force with the damping and biasing forces

$$m\,\frac{d^2y}{dt^2} = -N\,\frac{dy}{dt} - by \quad \ldots\ldots\ldots\ldots [21]$$

or

$$\frac{d^2y}{dt^2} + \frac{N\,dy}{m\,dt} + \frac{b}{m}\,y = 0 \quad \ldots\ldots\ldots\ldots [22]$$

for which a particular solution is

$$y = y_0 e^{-(N/2m)t} \cos \omega t \quad \ldots\ldots\ldots\ldots [23]$$

the angular velocity being

$$\omega = \sqrt{\left(\frac{b}{m} - \frac{N^2}{4m^2}\right)} \quad \ldots\ldots\ldots\ldots\ldots [24]$$

and the period of the damped motion, in terms of the undamped period T, is

$$T_d = \frac{2\pi}{\omega} = \frac{1}{\sqrt{\left[1 - \left(\frac{N^2}{4mb}\right)\right]}} T \qquad [25]$$

Equation [23] may represent three types of movement, depending on whether the radical of Equation [24] is positive, zero, or negative as determined by the degree of damping.

When

$$N < 2\sqrt{(mb)} \qquad [26]$$

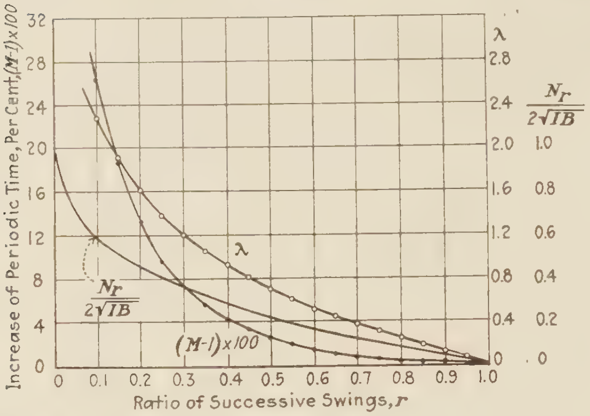

FIG. 5 THE RELATION BETWEEN THE DAMPING RATIO r AND λ, $N_r/2\sqrt{(IB)}$, AND $(M-1) \times 100$

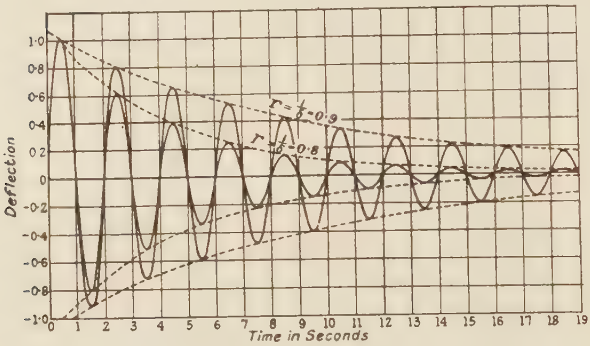

FIG. 6 OSCILLATING SYSTEMS WITH UNDAMPED PERIODIC TIME OF 1 SEC AND A DAMPING RATIO r OF 0.9 AND 0.8

the indicator oscillates with ever-decreasing amplitude according to Equation [23].

When

$$N = 2\sqrt{(mb)} = N_c \qquad [27]$$

then $\omega = 0$, $\cos \omega t = 1$, and the system is critically damped and the indicator asymptotically approaches its final position without oscillations. As will be seen later from Fig. 7, the critically damped return is substantially complete in the period T of one complete undamped oscillation.

When

$$N > 2\sqrt{(mb)} \qquad [28]$$

the system is over damped, and the indicator reaches the final position asymptotically but with greater delay.

It is useful in the analysis of damped harmonic motion to determine the ratio of amplitude of successive swings as

$$r = y_2/y_1 \qquad [29]$$

or the decrement

$$\delta = y_1/y_2 = 1/r \qquad [30]$$

and from the logarithm of the ratio of successive amplitudes we determine the logarithmic decrement

$$\lambda = \log_e (y_1/y_2) = \log_e \delta = \log_e (1/r) = -\log_e r = -NT_d/2m \qquad [31]$$

As was previously noted, the undamped period is

$$T = 2\pi\sqrt{(m/b)} \qquad [12]$$

With damping, the period is changed so that

$$M = T_d/T = [1 + (\lambda^2/2\pi^2)] \qquad [32]$$

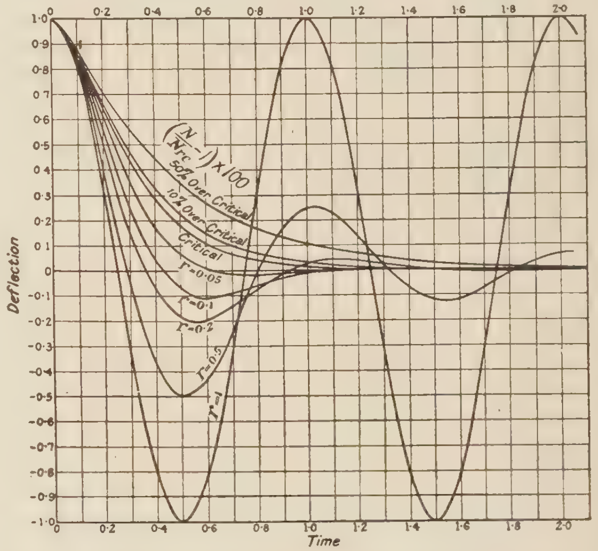

FIG. 7 CURVES OF OSCILLATING SYSTEMS WITH VARIOUS DEGREES OF DAMPING

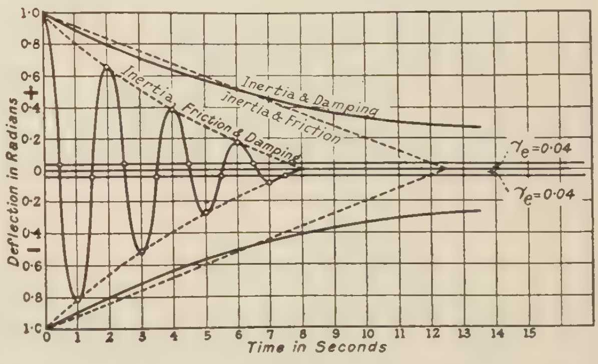

FIG. 8 OSCILLATING SYSTEM WITH INERTIA, FRICTION, AND DAMPING

Fig. 5 shows the increase of the periodic time, the logarithmic decrement, and a function $N/[2\sqrt{(mb)}]$ (used in computation), all plotted against the ratio of successive swings r.

Fig. 6 is self-explanatory.

Fig. 7 illustrates damped movements suitable for most instruments. For the greatest promptitude of accurate reading (at from 1 to 0.1 per cent of the deflection), a damping ratio of $r = 0.02$ is desirable; in other words, the damping constant N should be about 80 per cent of N_c.

Inertia, Damping, and Friction. As shown in Fig. 8 the

oscillations in this case suffer a further decrease in amplitude with successive swings due to the friction as well as the fluid damping. The graph illustrates the behavior of a system having $T = 1$ sec, $r = 0.9$, a friction dead-zone angle $\gamma_e = 0.04$ radian, and an initial deflection $\gamma_0 = 1$ radian.

Fig. 9 shows curves similar to those of Fig. 7 with a dead-zone angle $\gamma_e = 0.01\,\gamma_0$ added. However, only the end 10 per cent of the movement is drawn. In general, instruments having some friction should be even more under-damped than those without it, allowing an over-shoot of from 2 to 3 per cent, corresponding to a damping constant N/N_c of from 0.70 to 0.75.

The motion of the indicator subjected to a linear increase of the variable is shown in Fig. 10 to swing alternately over and under its proper value. Of course, in any practical instance, this superimposed harmonic motion would soon die out (3, 4). In any case the over and under errors offset each other almost exactly, except where there is a wide departure from a linear relation between the variable and the indicated function, as, for example, with a flow meter in which the pressure differential increases in a lineal manner; however, as the discharge varies as the square root of this differential, the meter would indicate too high a discharge on the average.

The motion of the indicator subjected to a quadratic increase of the variable has the indicator lag behind the proper value at all times as long as the variable is so increasing, which of course it cannot do indefinitely. It may be noted that the quadratic law accompanies a uniform acceleration of the variable as when a constant force is applied to any mass, or when a constant pressure is applied to a body of liquid in a (theoretically) frictionless line of considerable length. (See Fig. 11.)

Where the variable has a definite natural period for any reason, it is desirable to have the period of the indicator of the order of at least ten times as long, where sudden changes of the variable are likely to occur as with engine indicators. This is at best a rough rule.

Classification of Instruments

The classifications given in Table 1 are from different angles. However, the several classes have in common the fact that they

TABLE 1 CLASSIFICATION OF INSTRUMENTS

(*A*) Operation:
- 1 Self-operated
- 2 Servo-operated:
  - (*a*) Null method
  - (*b*) Booster method
- 3 Mixed, e.g., self-operated but servo-controlled

(*B*) Mode of operation:
- 1 Continuous
- 2 Cyclical

(*C*) Function:
- 1 Differentiating
- 2 Indicating or recording
- 3 Integrating
- 4 Regulating, that is, governing a controller (see *D*2 below)

(*D*) Complexity:
- 1 Simple, a single variable function
- 2 Compound, a plurality of variables in combination

(*E*) Auxiliaries:
- 1 Telemetering
- 2 Regulating:
  - (*a*) Same variable
  - (*b*) Other variables, in constant or predetermined ratios:
    - (*1*) Single variable
    - (*2*) Plurality of variables

(*F*) Principle ("indication" corresponding with the variable):
- 1 Length, either of distance or time
- 2 Force, without significant lineal departure
- 3 Pressure
- 4 Volume

are all taken from the standpoint of instruments *per se* instead of from the standpoint of their application or factors, such as cost, influencing this application. Each of these various classes is mutually exclusive, which is one criterion of consistency for classification. To be complete, it would be necessary to extend these from the standpoint of the application of the instruments in practice.

Analysis of Functioning of Instruments. The following six

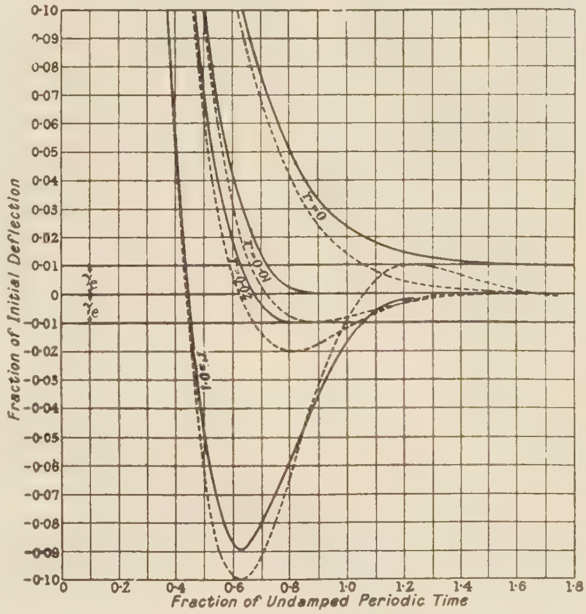

Fig. 9 System With Damping and Friction

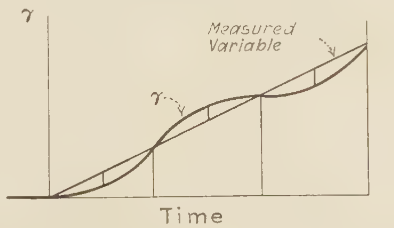

Fig. 10 Motion of the Indicator Upon a Lineal Increase of the Variable

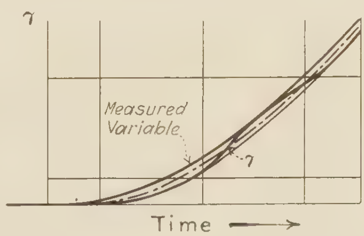

Fig. 11 Motion of the Indicator Upon a Quadratic Increase of the Variable

items should be used in analyzing the functioning of instruments:

1 Number of operations—directly or magnetically operated pen or counter.

2 Length—the usual basis of indication or recording.

3 Force—converted into length by stretching a spring, moving a weight to exert a different moment, displacing a volume of liquid, or opposed by an electro-magnet.

4 Pressure—converted into force by acting on an area.

5 Temperature—converted into length by lineal expansion, by volumetric expansion in turn by expanding an element having an area, or by a corresponding pressure change in turn by an

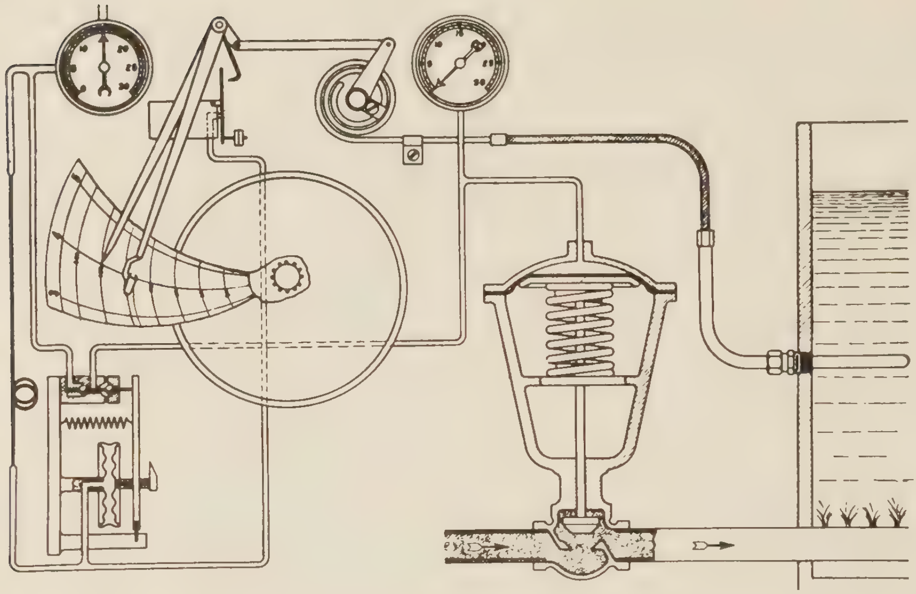

Fig. 12 Self-Operated Temperature Recorder With an Auxiliary Air-Operated Controller

area to a force and thence to a length, as before, by thermo-electric and electro-magnetic means.

6 Time-duration impulses—converted into length by a clock motor.

After obtaining a length corresponding to the variable, the instrument normally either indicates some function of this directly (or records it on a time chart), integrates some function of this length, or differentiates it. A recording circular chart may be rotated according to the variable and either time or another quantity recorded on the chart; for example, pressure on a flow chart. However, the chart is conventionally rotated at a constant speed and the variable is recorded on it. A formerly common type of integrator has a roller driven by a constant-speed friction disk at a variable radius corresponding with the variable, the roller commonly actuating a counter. This has been generally replaced by cyclically operated integrators now that electric motors are available to operate the latter as frequently as is required in practice, making their operation in effect continuous.

With the null method, a length (or counter reading) is set up according to a balance of forces or distances. In the steam shunt meter, a water-damped rotor drives a counter in direct ratio to the total flow. Similarly, in the case of a centrifugal integrator, in which the speed of a governor shaft is proportional to the square root of a differential pressure, a revolution counter indicates flow directly. These are illustrations of self-operated and servo-operated instruments. An electrical (or mechanical) tachometer may be used with either: this amounting to a differentiating type of meter.

With respect to attempts to classify instruments according to the variables measured, these seem to be less desirable than the foregoing classification and analysis of instruments according to their functioning. It is particularly futile to try to classify the variables on a dimensional basis; for example, L^2/T is either for kinematic viscosity or area per second and FL is either for torque or work.

Several examples of various classes of instruments are offered in the following paragraphs. These were chosen purely on the basis of their suitability as illustrations.

Example No. 1. Self-Operated Instrument. Fig. 12 shows a recorder controller having a closed-tube fluid-filled system in which the bulb is exposed to the temperature of liquid in a tank. The expansion of the fluid in the bulb, or its increase of pressure due to a temperature rise, for liquid and gas-filled systems or for vapor-pressure systems, respectively, is transmitted through

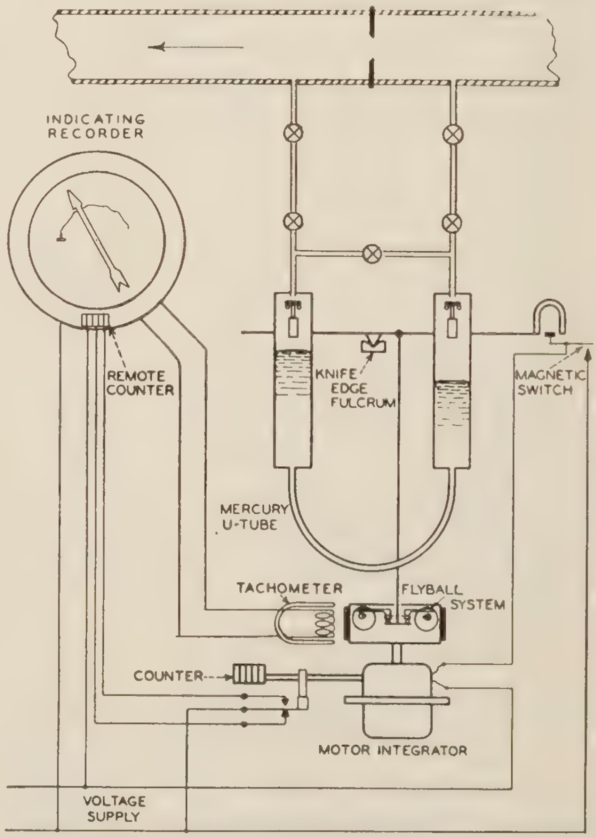

Fig. 13 Servo-Operated Null-Method Mechanical-Electrical Flow Meter

(Courtesy of the Instruments Publishing Company and the Leeds & Northrup Company.)

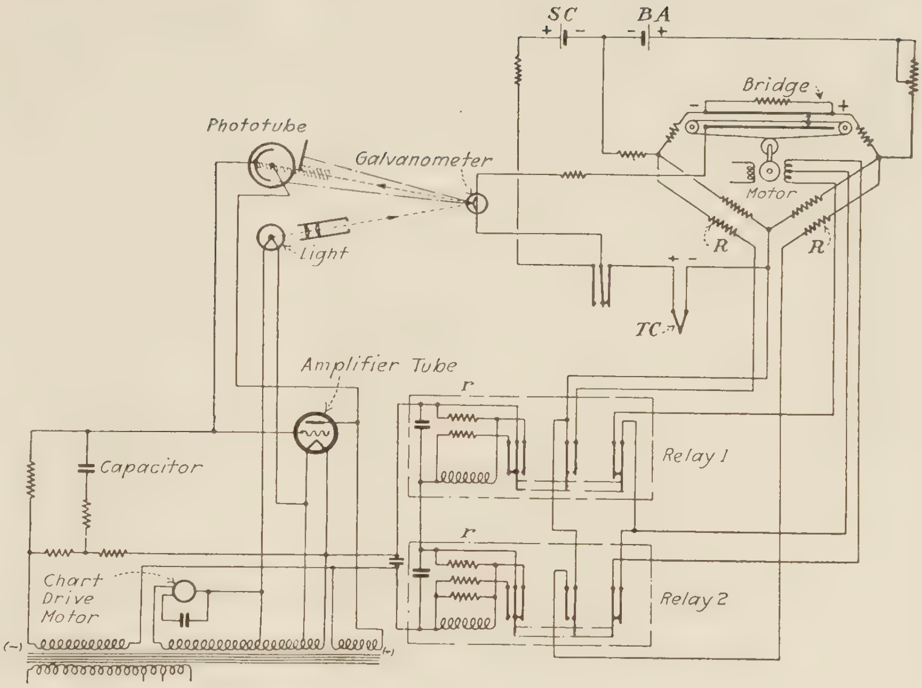

Fig. 15 Servo-Operated Null-Method Photoelectric Potentiometer Used to Measure Temperature

the capillary tubing to a Bourdon-type tube which uncoils slightly, by a corresponding amount. This moves the upper end of the flapper to the left so that its lower end clears the end of the nozzle, thus permitting air to escape and correspondingly altering the position of the air valve and consequently the air pressure in the diaphragm top of the valve for the supply of the heating fluid to the bottom of the tank. Thus, not only does the position of the recording element correspond with the temperature, but so does the air pressure in the line to the nozzle, the position of the air valve, the pressure in the line to the diaphragm top, and the position of the diaphragm top, since this is opposed by a spring. In this combination, the bulb may be termed the primary or sensitive element and the Bourdon tube the secondary or indicating element. Thus, the air pressure acts as a follow-up and the matter of sensitivity becomes a pertinent one.

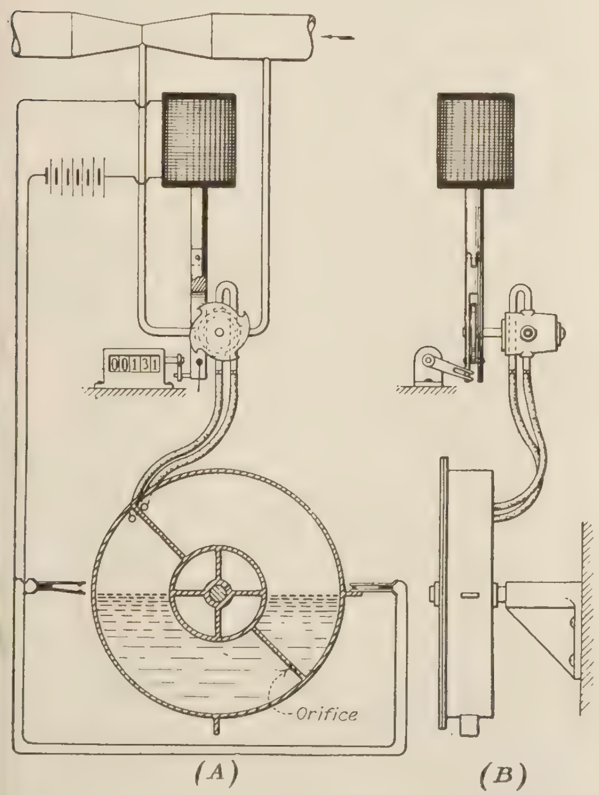

Fig. 14 Self-Operated Flow Meter of the Mixed Hydraulic-Electric Type

Example No. 2. Servo-Operated, Null-Method, Mechanical-Electrical. In Fig. 13 is shown a flow meter in which the force produced in a balanced U-tube, by a pressure differential accompanying flow through an orifice in a pipe, is opposed by the centrifugal force of a flyball system rotated at a speed governed by the position of the U-tube. This illustrates a system that is intermittently supplied with power at short intervals as are required, rather than in regular cycles, as to be substantially continuous in its action. The tachometer is a differentiating device for the revolutions of the flyballs while the counter is an indicator for them and hence for the total quantity of flow since a substantially quadratic relation exists between head and discharge and also between centrifugal force and speed of rotation. Further, this device also telemeters both the discharge and total quantities to a receiving instrument.

Example No. 3. Servo-Operated, Mixed Hydraulic, and Electric. In Fig. 14 are shown the front and side elevations of a flow meter illustrating this mixed type. The discharge through a venturi tube creates a differential pressure which is carried through a solenoid-actuated reversing valve to a balanced ring-type U-tube, in two legs of which the manometric liquid is separated by a wall containing a small orifice. As the manometric liquid flows from one leg to the other, the ring-balance oscillates from one extreme position to the other, thus operating by its limit switches the reversing valve and its associated counter. This is like the null-

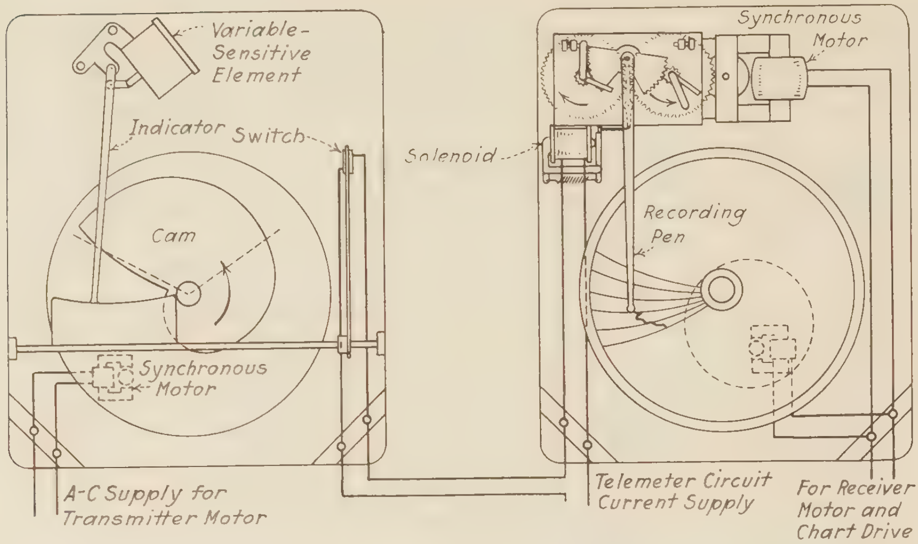

Fig. 16 Servo-Operated Booster-Method Mechanical Electrical Impulse-Duration Telemeter
(Courtesy of the Bristol Company.)

method flow meter in that the quadratic relation is automatically rectified without requiring a cam or other multiplying means, thus giving a longer operative range than with more conventional designs. This device also may act as a telemeter for the integrator by impulses spaced correspondingly with the speed, which impulses may consequently be used to govern an indicator for the discharge, i.e., a differentiating device.

Although this device may not seem offhand to be either a null-method or a booster-type, it actually is a null-method type in which the contacts are widely spaced but still keep the response within the set limits corresponding with a unit quantity so that counting impulses are telemetered in proportion to the actual total quantity discharged. If the ring-balance directly actuated the counter, while retaining the reversing switches for governing the reversals of the servo-operated valve, such a device could be classified as a mixed self-operated servo-governed type. Even though not commercially available at present, this device is included to emphasize the difficulties to be overcome in setting up a universally acceptable classification. (Note: Fig. 14 is from U. S. patent 2,077,444 to Walker.)

Example No. 4. Servo-Operated, Null-Method, Photoelectric. Recording temperature is often of vital importance to mechanical engineers in the process industries, e.g., in refining petroleum, and consequently a complete treatment of industrial instruments should include potentiometers used for this purpose.

Fig. 15 shows the electrical circuits of a potentiometer used with a thermocouple to record temperature. Although this instrument (5) is photoelectrically actuated, still its solution of the mechanical problem of balancing accurately at high speed is interesting. The problem of stopping a moving light beam on the edge of a photoelectric cell without any tendency to oscillate or hunt, and yet with speed and precision, has been solved by (1) applying an advancing electromotive force to the galvanometer while it is deflected, (2) using separate potentiometer and electronic circuits which are functionally related by electro-magnetic relays, and (3) delaying action at the far edge of the dead zone, when entered by the light beam, by delaying the response of the amplifier at the balance point. These improvements make possible balancing within a few hundredths of 1 per cent at a speed corresponding to 25 sec for travel over a 10-in. scale. At higher speeds, the slight error of balancing is proportionately increased.

The balancing power circuit includes only a reversing motor and the control contacts of the two relays shown in Fig. 15. With both relays closed, and both relays open, the reversing motor moves the slide-wire contact down scale and up scale, respectively. With relay No. 1 closed and relay No. 2 open, the motor stops.

The potentiometer circuit, has, in addition to its bridge circuit, two biasing resistances R which are so connected therewith as to apply the aforementioned advancing electromotive force in the proper direction whenever both relays are open or closed. At the balance point, relay No. 1 being closed and relay No. 2 being open, connection is made to the point corresponding to the compensated zero of the scale.

The galvanometer has a dead zone which corresponds to the deflection of the light beam required to raise the plate current from the opening value of relay No. 1 to the closing value of relay No. 2. The dead zone is only about one fifth of the width of the light beam at the controlling edge of the phototube.

Since this instrument may be at a distance from the thermocouple, it may be said to be telemetric in the same sense that any electrical instrument is telemetric, i.e., as not using a mechanical or hydraulic connection for the transmission of an indication.

However, reference to the classification in Table 1 will bring out the point that such a definition is objectionable. It seems preferable to require that a telemetric device utilize another system than that which is sensitive to the variable; for example, air pressure may be set up corresponding with a temperature.

Example No. 5. Servo-Operated, Booster Method, Mechanical. In Fig. 16, the motor-driven transmitter creates cyclical impulses of durations corresponding with the variable, the indicator of the transmitter being free during a portion of its cycle to assume a new position corresponding with a new value of the variable following a change in its magnitude. At the receiver, a similar motor drives a differential mechanism at constant speed. The impulses are there each converted by solenoid-operated clutches into angles corresponding with the variable.

In the case of an integrator (not shown), these angles are progressively added, while in the case of an indicator or recorder, these angles each start from a fixed point and a complementally moving arm moves through an angle measured from another fixed point and in the opposite direction during the interval between impulses. The result is that a braked recording pen arm is positioned in either direction, as required, by a separate source of power from that of the variable. The integrator is usually more accurate than the position of the braked pen arm since the two positioning arms for the latter must either have a slight play,

or back lash, otherwise the pen is likely to be nudged slightly in both directions in each cycle, even though the variable remain momentarily constant.

Although the gain in accuracy of a booster-type instrument over that of a self-operating instrument for the same variable may be open to question, still the flexibility of the former in regard to telemetering, integrating, and controlling is apparent. Also, the cam may be readily shaped to rectify any nonlineal function; for example, the square root for a flow meter.

Example No. 6. Compound Instruments. Cyclically operated instruments lend themselves to the integration of the product of a number of variables, as for example, flow meters for gas where it is commonly necessary to integrate both differentials and absolute pressures, and occasionally temperatures as well, the square roots being removed either before or after taking their product. Other cases are: The watt meter, obtaining the product of volts by amperes by the power factor; the belt conveyor weigher, totaling the product of belt speed by weight per unit length of belt; and ratio meters which compare steam flow with air flow.

Elements of Industrial Instruments

For pressures, the variable-responsive element is usually a Bourdon tube, a bellows, a flat or corrugated diaphragm, or a bell or a displacer, float, or even a bucket. These are opposed generally by a spring, weight, or other means for predetermining the sensitivity of the combination (b' of Equations [2] and [3]). The effective area of a flat diaphragm varies rapidly with its travel and so requires compensation for accuracy. The metal selected for any element which is deformed in measuring, should be selected to have minimum hysteresis; in general, the design should be such that creep near the ultimate value is made as small as practicable by the use of an element of sufficient length so the maximum unit deformation will not cause the metal to be stressed more than a small fraction of the proportional limit.

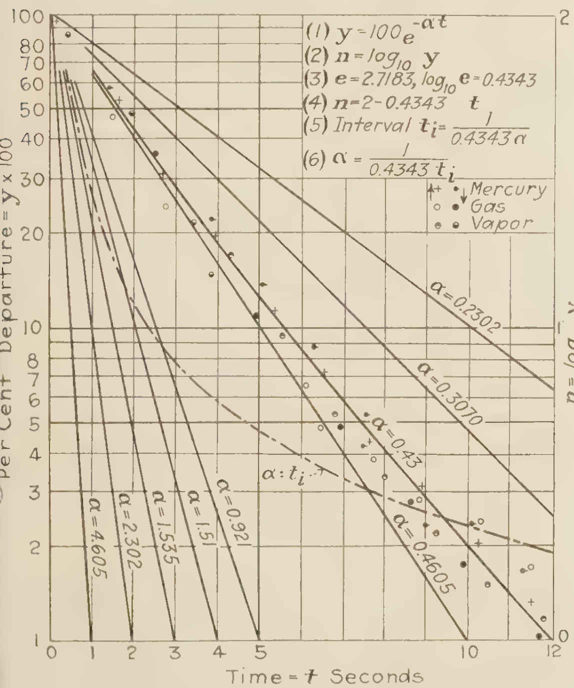

Fig. 17 Interval of Temperature-Responsive Fluid-Filled Conventional-Tube Systems

Where the pressure of a gas is being measured and pulsations exist, it is desirable to have a minimum volume of gas in the meter and to damp the pulsations in the pressure pipe with viscous flow through a capillary instead of through a throttling orifice; however, if the latter be used, great care must be taken that it be perfectly symmetrical. For measuring the pressure from a modern, high-speed rotary piston pump for liquids it is desirable to avoid the use of a pulsation deadener in the pressure pipe and, instead of a Bourdon tube or bellows, to use a weighted piston of the deadweight type which has enough inertia not to be appreciably disturbed by these high-frequency vibrations.

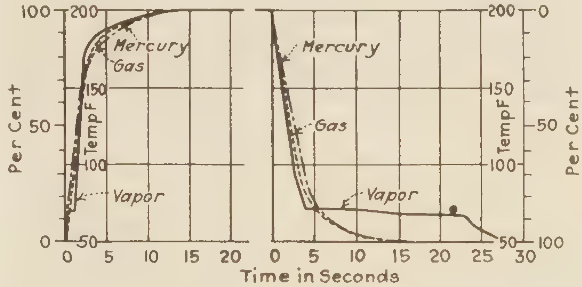

Fig. 18 Comparative Performance of Mercury, Gas-Type, and Vapor-Type Tube Systems

For temperature measurement, sealed systems (6, 7) are widely used in which a bulb, a connecting tube, and a bellows, or some other expansible element, responsive to volume or pressure changes, are filled with a gas, a liquid (completely full), or partly with a liquid which creates a vapor pressure corresponding with the temperature. The physical state, or phase, of the fluid used may slightly affect the speed of response of the system to changes in the temperature. The speed is conveniently measured by the "interval" required following a change for the variable to come within 10 per cent of the change from its final value, so that in twice the interval, the variable would be within 1 per cent of the change from the final value. In Fig. 17 (data by Paul F. K. Erbguth who is associated with the authors) is shown the performance of several different fluids in such tube systems, the data for the vapor type being shown thereon only where there is a proper difference of temperature between bulb and connecting tubing. The interval depends primarily upon the bulb design in regard to insulation and exposure to turbulence in a line, where flow can be used to scour away the film on the outside of the tube, fins being commonly added where space is available. The phase of the fluid used also affects the suitability of a given thermometer for a particular service, multiple bulbs on a common bellows responding to the average temperature with either gas-filled or liquid-filled systems, and the maximum temperature with the vapor system. Fig. 17 also shows the relation between α and the interval t_i plotted as a dot-and-dash line. Fig. 18 shows the test data of Fig. 17 more fully and conventionally plotted, bringing out the erratic performance of the vapor-type whenever the bulb temperature is within a few degrees fahrenheit of the ambient temperature. The bimetallic or metal expansion-stem element gives considerable force so that unexpected sensitivity may sometimes be obtained with these crude-appearing devices.

Flow meters use the differential-pressure, or metering head to indicate the flow rate or discharge. To obtain the square root and to bring out the movement of a float from its U-tube through a pressure wall are special problems for these meters. The formed well, formed bell, plain bell with formed displacer, formed cam with cylindrical or slightly coned wells, tilting U-tube with weighted cam or the use of electrical solenoids to oppose the tilt

are a few of the conventional solutions of the square-root problem. The cyclically operated cam can also be used for this purpose, especially where telemetering or controlling are required. The pressure-wall problem has been solved in various ways by the different manufacturers: The use of very small shafts with extremely small clearances, the use of slightly larger clearances with grease-packing, the use of plain and flexible "hat" packings on a larger shaft either with or without grease, the use of small flexible tubes on the tilting U-tube and the use of a magnetic float on the inside of a non-magnetic tube, or the use of a rotary magnetic coupling. Where a power-shaft is driven by an electric motor, to actuate a cam within the pressure chamber, it is apparent that this is still another solution of the same problem.

As discussed earlier in this paper, where metal is deformed it is desirable to keep the amount of deformation as small as possible relative to the actuating force. In any case, the meter should be balanced with zero differential exactly as it is at its lowest operat-

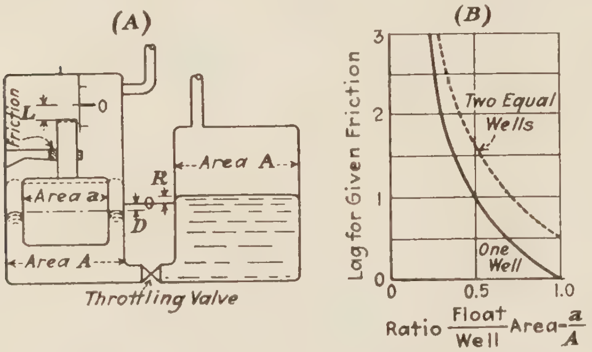

Fig. 19 Effect on Lag of Float Displacement and Throttling

ing rate. Further, the torque available must be adequate to give the required accuracy at low rates. It should be mentioned that only the tilting U-tube has an unlimited travel possible at low heads. All other forms require multiplication by one means or another, although the formed bell can be counterbalanced to be unstable near its zero position so that this form practically shares this advantage of long range. The other forms all follow the first power instead of square root at low heads.

Fig. 19 illustrates an interesting manometric phenomenon which is apparently not as widely known as its importance in flow metering requires: The narrowing of the dead zone, as the manometric liquid is throttled and as the float size approaches that of its well. Fig. 19*A* shows diagrammatically a manometric U-tube having both legs of equal area, one leg containing a float and a source of constant friction in the steam guide, an adjustable throttling valve being provided between the two legs. Fig. 19*B* has as its ordinate the variation of lag with the well, not containing the float, respectively connected and disconnected to indicate the possible limits of throttling, and as its abscissa the ratio of the float area to well area. This effect is due to the displacement of the float in its well. The practical result is that too much throttling of the manometric liquid gives a fictitious momentary liveliness to the float, but the level will gradually drift to a new value. This phenomenon has led many to consider that the ring-balance has a smaller dead zone than its ultimate value. The bell-type manometer must have the necessary damping in its connecting lines since obviously there can be no effective throttling of its manometric liquid.

The balancing of instruments sometimes presents unexpected difficulties for reliable operation, especially where the utmost range must be obtained with an instrument such as a flow meter in which the necessary multiplication increases as the forces available decrease. In some cases, the use of light metals and tubular sections is indicated. Where cams are used, the cam roller should be at least three times the size of its axle and thought should be given as to the best location of centers and whether an inside or outside cam is preferable; also, the force holding the roller against the cam should be enough to ensure the rotation of the roller and thus prevent its wearing flat in one spot.

It is generally preferable to have a single part or unit system for a single function in instruments, except of course such parts as mounting brackets which must serve to hold the several parts in their operating relations. A discussion of electrical elements is omitted as being outside of the scope of the paper.

Conclusion

Notes on instruments generally, including manufacture, shop and field calibrations, and methods of adjustment, belong in any complete treatment of this subject. Examples of these are:

1 To make instruments available commercially, production methods may require working to tolerances that eliminate adjustments in final assembly. Close tolerances are essential in manufacture, and preassembly and subassembly inspection is desirable. Inventory control and an efficient balance between the cost of labor tied up in placing instruments in stock and that done on the final order frequently have more to do with the availability of instruments on an economic basis than their basic design.

2 In calibrations and especially in adjustments, the use of rule-of-thumb methods is objectionable; it is preferable to make a measured change and then measure the effect of such change. In such work it generally saves time to follow the principle used in spotting artillery fire, that is, first fire over, then under, and then correct.

3 The present treatment is limited to variable quantities. In these it is difficult to lay too much emphasis on rate of change, clock motors, and the laws of variation of damping of various sorts. The excellence of small modern electric motors has opened up an entirely new era in instruments and has been the prime factor leading to the general acceptance of servo-operated instruments. However, it must be noted that such motors are not ordinarily acceptable in explosive atmospheres, since even where there are no open contacts the insulation of the small wires has been known to break down and in some cases the wires break from vibration even when apparently anchored securely. Where an explosion-proof instrument is to be servo-operated, such motors should be sealed in an explosion-proof housing or be submerged under a nonflammable liquid of low vapor pressure and the container preferably be sealed against the air. This requirement may seem harsh, but human life and property must not be placed in jeopardy by instruments.

4 The adjustment of cyclically operable instruments, or any instrument or regulator after installation on a process having a natural long-time cycle of its own, may require an unexpectedly large amount of time. In one instance, an instrument worked correctly over the required range in a shop test and in a field calibration, but occasionally failed in the middle of the night. It took an all-night vigil to discover that there was an interference with the instrument case when the variable reached a given high value just before dawn. While an instrument engineer may not appear to be working, but to be merely watching an instrument operate, it must be remembered that the instrument can only move at a fixed rate regardless of the impatience of the observer.

5 On the other hand, it should be noted that models have a universal appeal to all who are in the least mechanically inclined, and instruments are essentially model machines. Also, they present fascinating problems, often requiring a subtlety of analysis, as when one must confidently act upon a mental picture of what

occurs behind metal walls, to reach a solution. Also, in meeting problems in this field, it must be remembered that: "This field has been plowed, harrowed, and cross-harrowed by successive inventors." The consequence is that there is occasionally an interesting patent phase to this work.

In conclusion, others are urged to improve on and expand this treatment. However, this paper is submitted so that open discussion may show the most promising course to be followed in a more extended treatment. The authors' first contacts with industrial instruments were as "users" and their aim is to lay the basis for a concise but thorough treatment of this subject which will be of the greatest benefit to those who use industrial instruments and regulators in the process industries.

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Abstract of Progress Report No. 3 on Heavy Helical Springs[1]

By C. T. EDGERTON,[2] NEW YORK, N. Y.

This abstract gives the principal results of static tension and torsion tests, rotating-beam endurance tests in bending, and torsion endurance tests of straight pieces of plain carbon basic open-hearth, plain carbon electric-furnace and silicon-vanadium spring steel. These results are compared with those for other steels.

THE RESEARCH program of the Subcommittee on Heavy Helical Springs of the A.S.M.E. Special Research Committee on Mechanical Springs, as described previously[3,4] in Progress Reports Nos. 1 and 2, was completely at a standstill for some time because of economic conditions. About a year ago, J. B. Johnson, chief of Materials Branch of the Air Corps at Wright Field, became interested in the Committee's work and offered to conduct some fatigue tests. Mr. Johnson had previously made some similar tests, the results of which were later published.[5]

Four groups of material were furnished for the new series of tests of which groups *A* and *B* were of plain carbon basic open-hearth steel, group *C* was of plain carbon electric-furnace steel, and group *D* was of silicon-vanadium steel.

All springs were of $^3/_4$ in. round steel with $7^1/_2$ total coils and, with the exception of group *B*, were of index 5. Group *B* was of index 3 and was intended, by comparison of fatigue results with those of index 5, to supply information as to the application of the Rover-Wahl stress correction as a measure of endurance. The chemical composition and heat-treatment given the four groups of steels are given in Table 1.

The springs in each group were coiled to finish at a graded series of free heights, and were fatigue tested at a similarly graded series of maximum stresses. A probable stress-endurance curve was then calculated for each group on the assumption that the curve is of the form

$$(N - a)(S - b) = k \qquad [1]$$

where S is the maximum test stress, N is the number of cycles of stress to failure, and a, b, and k are constants for the group.

Preliminary and calibration tests which were made were substantially in accordance with the tentative program drafted by the Subcommittee at the outset of its work. Mr. Johnson made static tension and torsion tests, rotating-beam endurance tests in bending, and torsion endurance tests—all on straight pieces of the same material as the springs. He also made magnaflux tests on the springs, all of which revealed no cracks whatever.

TABLE 1 CHEMICAL COMPOSITION AND HEAT-TREATMENT GIVEN THE FOUR GROUPS OF STEELS

	Groups *A* and *B*	Group *C*	Group *D*
Chemical composition:			
Carbon, per cent	1.050	1.050	0.940
Manganese, per cent	0.360	0.210	0.460
Phosphorus, per cent	0.020	0.008	0.019
Sulphur, per cent	0.032	0.011	0.014
Silicon, per cent	0.150	0.270	0.630
Vanadium, per cent	0	0	0.170
Heat-treatment:			
Oil-quench temperature, F	1620	1640	1570
Draw temperature, F	700	700	850–890
Time, hr	1	1	1
Cooling medium	Air	Air	Air

Note: Groups *A* and *B*—Plain carbon basic open-hearth steel.
Group *C*—Plain carbon electric-furnace steel.
Group *D*—Silicon-vanadium steel.

Results of the static tests on the springs are given in Tables 2 to 9, inclusive. Results of the endurance tests on the springs are shown in Figs. 1 to 4, inclusive. Results of the tests on the straight pieces are given in Table 10. A comparison of Mr. Johnson's results, as previously published,[5] with the results on the Committee's springs are given in Table 11.

Comparing the results in Table 11, the plain carbon electric-furnace steel used in Mr. Johnson's tests is far better than all other grades tested. These springs were reported to be cold-coiled and there were indications that the steel had been cold-drawn. These factors may or may not have had some bearing on the results, and will be given consideration in the Committee's plans for further work.

Next best, by a small margin, is the electric-furnace chrome-vanadium steel, followed by the plain carbon basic open-hearth steel. The springs of plain carbon electric-furnace steel furnished by the Committee were probably substandard in regard to both chemical composition and heat-treatment. This group, as well as the silicon-vanadium steel, rated rather low in the comparison.

It should be remembered that all the springs originally tested by Mr. Johnson were fabricated from centerless ground bars, whereas the Committee's springs were coiled from bars "as-rolled." Apparently this has not resulted in any enormous difference in endurance, such as is often found in rotating-beam

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[1] Abstract of Progress Report No. 3 of the Subcommittee on Heavy Helical Springs of the A.S.M.E. Special Research Committee on Mechanical Springs.

[2] Head of Bureau of Statistics, Crucible Steel Company of America. Mr. Edgerton was graduated from Cornell University with a B.S. degree in mechanical engineering in 1901 and then entered the employ of the Brooklyn Rapid Transit Company. From 1902 to 1915 he was associated in various capacities with the Railway Steel-Spring Company, part of this time being devoted to research work. Since 1915 he has been connected with the Crucible Steel Company of America. He has served for several years on the A.S.M.E. Special Research Committee on Mechanical Springs as secretary of the committee, chairman of the Subcommittee on Heavy Helical Springs, and Society representative of the American Society for Testing Materials.

[3] "Fatigue Tests of Helical Springs and Number of Inactive Coils in Compression," by C. T. Edgerton, Progress Report No. 1 of the Subcommittee on Heavy Helical Springs of the A.S.M.E. Special Research Committee on Mechanical Springs, presented at the Annual Meeting of The American Society of Mechanical Engineers, held in New York, N. Y., December 5 to 9, 1932.

[4] Progress Report No. 2 of the Subcommittee on Heavy Helical Springs of the A.S.M.E. Special Research Committee on Mechanical Springs, by C. T. Edgerton, presented at a meeting of The American Society of Mechanical Engineers, held in Chicago, Ill., June 27, 1933.

[5] "Fatigue Characteristics of Helical Springs," by J. B. Johnson, *Iron Age*, March 15 and March 22, 1934.

The complete paper from which this abstract was taken was contributed by the Subcommittee on Heavy Helical Springs of the A.S.M.E. Special Research Committee on Mechanical Springs, and was presented at the Annual Meeting of The American Society of Mechanical Engineers, held in New York, N. Y., November 30 to December 6, 1936.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1937, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

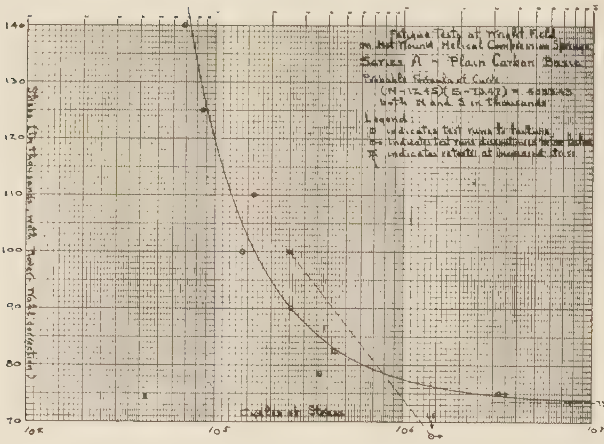

Fig. 1 Johnson's Results of Fatigue Tests of Hot-Wound Helical Springs of Plain Carbon Basic Steel—Group *A*

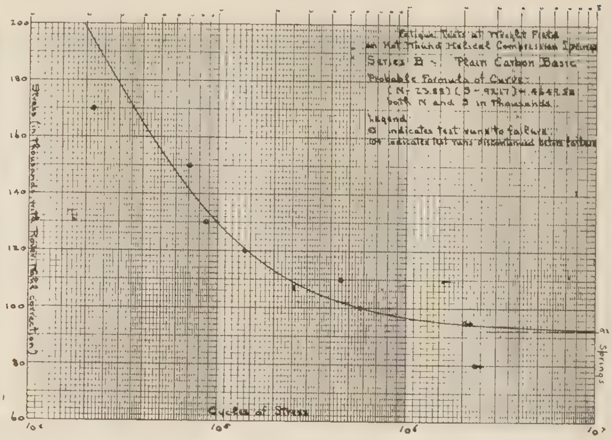

Fig. 2 Johnson's Results of Fatigue Tests of Hot-Wound Helical Springs of Plain Carbon Basic Steel—Group *B*

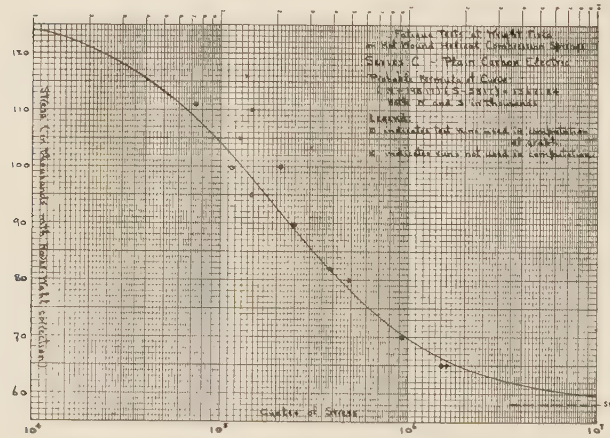

Fig. 3 Johnson's Results of Fatigue Tests of Hot-Wound Helical Springs of Plain Electric-Furnace Steel—Group *C*

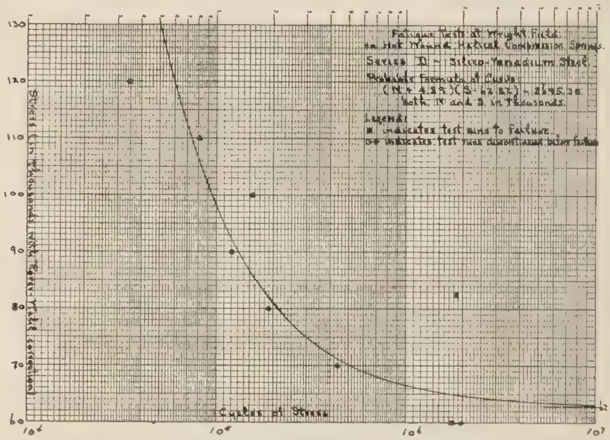

Fig. 4 Johnson's Results of Fatigue Tests of Hot-Wound Helical Springs of Silicon-Vanadium Steel—Group *D*

tests. We should not expect to find such a wide difference, as the various heating operations on the springs, subsequent to the centerless grinding, make it fairly certain that the finished article is neither bark-free nor surface-perfect. However, it is probable that the centerless grinding has contributed a measure of superiority to the springs so processed.

Coming now to a comparison of the tests on the two plain carbon basic groups of index 3 and 5, respectively, we find most interesting data. The endurance limit for the index-5 group, as given in Table 11, was 72,730 lb per sq in., while that of the index-3 group was 92,170 lb per sq in. On translating these figures into stresses figured by the conventional spring formula, without the Rover-Wahl correction, we get 56,450 lb per sq in. for the index-5 group, and 59,230 lb per sq in. for the index-3 group. It would appear from this comparison that the stress augment due to curvature of the bar has no effect whatever on endurance.

This is a most surprising conclusion, and further check tests are imperative. Some spring engineers, arguing from practical experience, have thought it probable that the Rover-Wahl correction overcorrects. However, there is no reason to doubt that the Rover-Wahl theory is sound; the stress augment it predicts is actually present in the spring, and it is difficult to believe that it has no effect on the endurance.

Mr. Johnson has kindly offered to conduct further tests for the Committee, and at the time this paper was written two additional groups of springs had just been forwarded to Wright Field. One of these is a repeat lot of plain carbon electric steel while the other is a group made of plain carbon acid open-hearth steel.

Our further plan is to select from all the grades of material so far tested the one which gives the best endurance results, and with this material as a standard proceed to make comparative tests for the effect of various surface finishes.

Progress Report No. Three. Section II

Springs Series A. Table II Calibration Tests.

Loads in Pounds	Overall Heights Compression Front	Back	Averge	Release Front	Back	Averge	Loads in Pounds	Overall Heights Compression Front	Back	Averge	Release Front	Back	Averge
Spring A-1.							Spring A-2.						
Free	8.751	8.947	8.849	8.696	8.891	8.794	Free	8.758	8.731	8.745	8.706	8.670	8.688
500	8.490	8.510	8.500	8.346	8.360	8.353	500	8.364	8.375	8.370	8.210	8.233	8.222
1000	8.120	8.114	8.117	7.882	7.878	7.880	1000	7.977	7.978	7.978	7.776	7.782	7.779
1500	7.738	7.723	7.731	7.457	7.446	7.452	1500	7.628	7.622	7.625	7.334	7.331	7.333
2000	7.353	7.339	7.346	7.007	6.999	7.003	2000	7.210	7.205	7.208	6.922	6.919	6.921
2500	6.952	6.938	6.945	6.627	6.616	6.622	2500	6.834	6.830	6.832	6.530	6.527	6.529
3000	6.532	6.520	6.526	6.229	6.218	6.224	3000	6.429	6.425	6.427	6.179	6.177	6.178
3500	6.120	6.108	6.114	5.854	5.844	5.849	3500	5.984	5.980	5.982	5.783	5.781	5.782
4000	5.666	5.654	5.660	5.473	5.463	5.468	4000	5.545	5.541	5.543	5.431	5.430	5.431
4500	5.227	5.214	5.221										
Solid	5.126	5.121	5.124				Solid	5.167	5.171	5.169			
Spring A-3.							Spring A-4.						
Free	8.649	8.601	8.625	8.618	8.542	8.580	Free	8.436	8.453	8.445	8.396	8.412	8.404
500	8.251	8.262	8.257	8.149	8.169	8.159	500	8.041	8.051	8.046	7.964	7.987	7.976
1000	7.867	7.871	7.869	7.718	7.725	7.722	1000	7.691	7.696	7.694	7.555	7.563	7.559
1500	7.487	7.484	7.486	7.298	7.292	7.295	1500	7.290	7.287	7.289	7.141	7.141	7.141
2000	7.097	7.093	7.097	6.880	6.877	6.879	2000	6.976	6.976	6.976	6.790	6.790	6.790
2500	6.710	6.706	6.708	6.502	6.500	6.501	2500	6.569	6.568	6.569	6.397	6.397	6.397
3000	6.317	6.314	6.316	6.116	6.114	6.115	3000	6.200	6.200	6.200	6.037	6.036	6.037
3500	5.908	5.906	5.907	5.744	5.742	5.743	3500	5.820	5.821	5.821	5.689	5.687	5.688
4000	5.507	5.504	5.506	5.372	5.370	5.371	4000	5.414	5.415	5.415	5.343	5.343	5.343
Solid	5.057	5.059	5.058				Solid	5.126	5.127	5.127			
Spring A-5.							Spring A-6.						
Free	8.432	8.453	8.443	8.348	8.414	8.381	Free	8.304	8.315	8.310	8.275	8.275	8.275
500	8.092	8.102	8.097	7.982	8.001	7.992	500	7.926	7.936	7.931	7.853	7.870	7.862
1000	7.738	7.743	7.741	7.579	7.588	7.584	1000	7.595	7.599	7.597	7.496	7.506	7.501
1500	7.366	7.361	7.364	7.163	7.163	7.163	1500	7.226	7.224	7.225	7.058	7.055	7.057
2000	6.987	6.983	6.985	6.805	6.804	6.805	2000	6.860	6.856	6.858	6.675	6.671	6.693
2500	6.606	6.602	6.604	6.427	6.426	6.427	2500	6.453	6.449	6.451	6.307	6.305	6.306
3000	6.213	6.210	6.212	6.050	6.048	6.049	3000	6.047	6.043	6.045	5.922	5.920	5.921
3500	5.827	5.824	5.826	5.703	5.701	5.702	3500	5.643	5.640	5.642	5.552	5.550	5.551
4000	5.430	5.427	5.429	5.357	5.358	5.358							
Solid	5.159	5.147	5.153				Solid	5.120	5.120	5.122			
Spring A-7.							Spring A-8.						
Free	7.520	7.436	7.478	7.512	7.418	7.465	Free	7.486	7.485	7.486	7.476	7.469	7.473
500	7.081	7.096	7.089	7.092	7.113	7.103	500	7.144	7.164	7.154	7.112	7.134	7.123
1000	6.774	6.785	6.780	6.753	6.766	6.760	1000	6.800	6.811	6.806	6.772	6.781	6.777
1500	6.423	6.425	6.424	6.379	6.381	6.380	1500	6.458	6.460	6.459	6.418	6.419	6.419
2000	6.058	6.058	6.058	6.027	6.026	6.027	2000	6.126	6.128	6.127	6.084	6.086	6.085
2500	5.703	5.701	5.702	5.686	5.685	5.686	2500	5.793	5.795	5.794	5.763	5.766	5.765
3000	5.354	5.353	5.354	5.338	5.338	5.338	3000	5.474	5.476	5.475	5.459	5.462	5.461
Solid	5.096	5.104	5.100				Solid	5.145	5.154	5.150			
Spring A-9													
Free	7.547	7.480	7.514	7.537	7.467	7.502	2000	6.139	6.139	6.139	6.112	6.111	6.112
500	7.187	7.203	7.195	7.147	7.164	7.156	2500	5.805	5.804	5.805	5.777	5.776	5.777
1000	6.835	6.842	6.839	6.808	6.812	6.810	3000	5.495	5.495	5.495	5.490	5.489	5.490
1500	6.484	6.484	6.484	6.457	6.456	6.457	Solid	5.222	5.226	5.224			

Progress Report No. Three Section II

Springs Series A Table II Page 2.

Deflection Readings on Four Active Coils.

Loads in Pounds	Spring A-1 Compression	Release	Spring A-3 Compression	Release	Spring A-9 Compression	Release
Free	5.65	5.60	5.63	5.61	4.74	4.72
500	5.48	5.34	5.40	5.31	4.54	4.48
1000	5.24	5.03	5.13	5.02	4.30	4.27
1500	4.97	4.76	4.88	4.73	4.06	4.03
2000	4.72	4.46	4.63	4.49	3.81	3.80
2500	4.49	4.21	4.40	4.21	3.60	3.58
3000	4.19	3.95	4.14	3.94	3.39	3.38
3500	3.90	3.68	3.86	3.70		
4000	3.61	3.44	3.54	3.40		
4500	3.30					
Solid	3.09		3.12		3.15	

Spring Measurements, and Development of Constants.

Spring No.	A-1	A-2	A-3	A-4	A-5	A-6	A-7	A-8	A-9
Size of Bar (1)	.736"	.737"	.729"	.728"	.733"	.731"	.731"	.735"	.738"
Outside Diameter (2)	4.581"	4.576"	4.555"	4.540"	4.534"	4.533"	4.498"	4.516"	4.511
Mean Helix Diameter (3)	3.781"	3.778"	3.761"	3.753"	3.743"	3.746"	3.723"	3.737"	3.727"
Spring Index (4)	5.137	5.130	5.159	5.155	5.106	5.124	5.093	5.088	5.050
Rover-Wahl Correction Factor	1.3010	1.3015	1.2995	1.2998	1.3031	1.3019	1.3040	1.3044	1.3070
M.F.S. per 1000-lb. Load — Conventional Formula	24149	24082	24722	24772	24204	24422	24272	24013	23615
M.F.S. per 1000-lb. Load — with Rover Wahl Corr.	31418	31343	32126	32199	31540	31795	31651	31323	30865
Capacity Solid - lbs.	4570	4410	4550	4330	4350	4140	3355	3480	3415
M.F.S. Solid — Conventional Formula	110360	106200	112485	107260	105290	101107	81430	83565	80645
M.F.S. Solid — with Rover Wahl Corr.	143580	138220	146170	139420	137200	131630	106190	109000	105400

(1) Average of six measurements.
(2) Average of six measurements, not corrected for pitch angle.
(3) Corrected for pitch angle (perpendicular to helical axis)
(4) Ratio ~ Mean Helix Diameter / Size of Bar.

Progress Report No. Three Section II

Springs Series A Table III. Tests at Wright Field

Series A Springs. Spring No.	Free Height	Solid Height	Total Deflection	Ratio Deflection / Solid Hgt.	Set in Inches — Static Loading, Compression No. 1.	2.	3.	Fatigue Loading, No. of Cycles 6000	10000	Total Set	Load in lbs. used in Fatigue Test	Stress Rover Wahl Formula	Number Cycles to Failure
1	8.73	5.16	3.57	.69	.05	.06	.00	.12	.00	.23	4460	140000	68500
2	8.68	5.19	3.49	.67	.06	.02	.02	.00	.00	.10	2507	78500	363400
3	8.56	5.06	3.50	.69	.05	.02	.02	.01	.00	.10	3890	125000	86400
4	8.37	5.13	3.24	.63	.05	.02	.01	.00	.00	.08	3416	110000	161000
5	8.35	5.21	3.14	.60	.06	.02	.03	.01	.00	.11	3178	100000	140700
6	8.25	5.16	3.09	.60	.05	.03	.01	.00	.00	.09	2835	90000	254000
7	7.44	5.14	2.30	.45	.05	.01	.00	.00	.00	.06	2615	82500	437600
8	7.46	5.18	2.28	.44	.02	.00	.00	.00	.00	.02	2408	75000	3228400 No failure
9	7.48	5.26	2.22	.42	.02	.00	.00	.00	.00	.02	2117	65000	1437300 No failure
9 (retest)	7.46	5.26	2.20	.42	.02	.00	.00	.00	.00	.02	3258	100000	258200 Retest

Brinell Hardness taken on flats cut from springs - 441 to 460

Progress Report No. Three. Section III

Springs Series B. Table IV Calibration Tests

Loads in Pounds	Overall Heights Compression Front	Back	Average	Release Front	Back	Average	Loads in Pounds	Overall Heights Compression Front	Back	Average	Release Front	Back	Average
Spring B-1.							Spring B-2.						
Free	6.542	6.606	6.574	6.523	6.586	6.555	Free	6.508	6.528	6.518	6.481	6.511	6.496
1000	6.400	6.390	6.395	6.325	6.321	6.323	1000	6.340	6.340	6.340	6.269	6.272	6.271
2000	6.241	6.226	6.234	6.125	6.115	6.120	2000	6.180	6.174	6.177	6.070	6.065	6.068
3000	6.062	6.050	6.056	5.925	5.915	5.920	3000	6.010	6.005	6.008	5.902	5.898	5.900
4000	5.898	5.887	5.893	5.749	5.738	5.744	4000	5.842	5.838	5.840	5.724	5.720	5.722
5000	5.714	5.702	5.708	5.578	5.567	5.573	5000	5.659	5.655	5.657	5.542	5.538	5.540
6000	5.522	5.511	5.517	5.417	5.407	5.412	6000	5.473	5.469	5.471	5.403	5.400	5.402
7000	5.316	5.306	5.311	5.260	5.250	5.255							
Solid	5.088	5.081	5.085				Solid	5.147	5.143	5.145			
Spring B-3							Spring B-4.						
Free	6.427	6.391	6.409	6.410	6.360	6.385	Free	6.098	6.038	6.068	6.088	6.034	6.061
1000	6.208	6.210	6.209	6.172	6.177	6.175	1000	5.883	5.877	5.880	5.858	5.860	5.859
2000	6.060	6.055	6.058	6.003	5.998	6.001	2000	5.739	5.730	5.735	5.690	5.688	5.689
3000	5.892	5.885	5.889	5.812	5.807	5.810	3000	5.581	5.574	5.578	5.539	5.535	5.537
4000	5.732	5.726	5.729	5.637	5.628	5.633	4000	5.400	5.396	5.398	5.369	5.365	5.367
5000	5.553	5.548	5.551	5.481	5.478	5.480							
6000	5.381	5.376	5.379	5.331	5.327	5.329							
Solid	5.075	5.069	5.072				Solid	4.956	4.958	4.957			
Spring B.5.							Spring B-6.						
Free	6.220	6.152	6.186	6.215	6.141	6.178	Free	6.172	6.183	6.178	6.158	6.182	6.170
1000	6.019	6.015	6.017	5.988	5.988	5.988	1000	5.911	5.907	5.909	5.981	5.986	5.984
2000	5.863	5.856	5.860	5.813	5.809	5.811	2000	5.861	5.857	5.859	5.808	5.804	5.806
3000	5.707	5.702	5.705	5.645	5.640	5.643	3000	5.704	5.702	5.703	5.653	5.649	5.651
4000	5.541	5.535	5.538	5.488	5.483	5.486	4000	5.548	5.544	5.546	5.500	5.496	5.498
5000	5.379	5.375	5.377	5.332	5.325	5.329	5000	5.385	5.380	5.383	5.338	5.336	5.337
Solid	5.037	5.036	5.037				Solid	5.077	5.078	5.078			
Spring B-7.							Spring B-8						
Free	6.064	6.055	6.060	6.058	6.048	6.053	Free	5.982	6.057	6.020	5.972	6.046	6.009
1000	5.888	5.893	5.891	5.880	5.888	5.884	1000	5.862	5.865	5.864	5.840	5.843	5.842
2000	5.738	5.736	5.737	5.712	5.708	5.710	2000	5.698	5.691	5.695	5.669	5.663	5.666
3000	5.585	5.580	5.583	5.555	5.551	5.553	3000	5.540	5.534	5.537	5.504	5.496	5.500
4000	5.430	5.425	5.428	5.401	5.397	5.399	4000	5.376	5.370	5.373	5.341	5.334	5.338
Solid	5.056	5.052	5.054				Solid	5.005	4.996	5.001			
Spring B-9.													
Free	6.002	5.987	5.995	5.988	5.981	5.985							
1000	5.820	5.823	5.822	5.806	5.810	5.808							
2000	5.674	5.668	5.671	5.651	5.647	5.649							
3000	5.514	5.508	5.511	5.493	5.487	5.490							
4000	5.353	5.347	5.350	5.345	5.339	5.342							
Solid	5.000	4.996	4.998										

Progress Report No. Three. Section III

Springs Series "B" Table IV Page 2.

Deflection Readings on Four Active Coils.

Loads in Pounds	Spring B-1. Compression	Release	Spring B-3 Compression	Release	Spring B-9. Compression	Release
Free	4.10	4.09	3.90	3.89	3.57	3.57
1000	4.01	3.96	3.78	3.75	3.49	3.46
2000	3.90	3.83	3.66	3.60	3.40	3.37
3000	3.78	3.69	3.56	3.47	3.28	3.26
4000	3.69	3.59	3.45	3.38	3.20	3.18
5000	3.57	3.48	3.33	3.30		
6000	3.47	3.36	3.21	3.17		
7000	3.33	3.26				
Solid	3.16		3.00		2.94	

Spring Measurements, and Development of Constants.

Spring No.		B-1	B-2	B-3	B-4	B-5	B-6	B-7	B-8	B-9
Size of Bar (1)		.746	.730	.736	.732	.732	.739	.732	.723	.729
Outside Diameter (2)		3.036	3.030	3.018	3.039	3.035	3.032	3.027	3.005	3.018
Mean Helix Diameter (3)		2.232	2.245	2.230	2.263	2.257	2.247	2.251	2.238	2.246
Spring Index (4)		2.992	3.075	3.030	3.092	3.083	3.041	3.077	3.095	3.083
Rover-Wahl Correction Factor		1.5820	1.5614	1.5725	1.5574	1.5596	1.5697	1.5610	1.5567	1.5596
MFS per 1000-lbs Load	Conventional Formula	13689	14696	14243	14693	14654	14177	14645	15081	14794
	with Rover-Wahl Correct.	21656	22946	22397	22883	22854	22254	22861	23467	23073
Capacity Solid - Pos.		8030	7680	7720	6480	7050	6870	6420	6295	6170
MFS. Solid	Conventional Formula	109920	112865	109960	95210	103310	97400	94020	94935	91280
	with Rover-Wahl Correct.	173900	176225	172900	148280	161120	152885	146770	147790	142360

(1) Average of six measurements.

(2) Average of six measurements, not corrected for pitch angle.

(3) Corrected for pitch angle. (perpendicular to helical axis)

(4) Ratio - Mean Helix Diameter/Size of Bar.

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Springs Series B Table V Tests at Wright Field

Spring No	Measurements Free Height	Solid Height	Total Deflect'n	Ratio Deflect'n / Solid Hgt	Set in Inches Static Loading No Compressions 1.	2	3	Fatigue Loading No. Cycles 6000	10000	Total Set	Load in lbs. used in Fatigue Test	Stress - Rover Wahl Formula	Number Cycles to Failure
B-1	6.54	5.09	1.45	.285	.02	.02	.01	.07	.02	.14	7859	170 000	21900
B-2	6.50	5.16	1.34	.260	.02	.01	.01	.03	.01	.08	5634	130 000	86400
B-3	6.39	5.11	1.28	.250	.01	.01	.01	.00	.00	.03	5332	120 000	138600
B-4	6.04	5.00	1.04	.208	.01	.00	.00	.01	.00	.02	4799	110 000	451800
B-5	6.15	5.08	1.07	.211	.01	.01	.01	.01	.00	.04	6520	150000	70300
B-6	6.14	5.10	1.04	.204	.01	.01	.00	.01	.00	.03	4493	100000	570400
B-7	6.04	5.09	.95	.187	.01	.00	.00	.01	.00	.02	4132	95000	580100
B-8	6.01	5.03	.98	.195	.01	.00	.00	.01	.00	.02	3749	80000	2331300 No Failure
B-9	5.96	5.03	.93	.185	.01	.01	.01	.00	.00	.02	4040	95000	2074500 No Failure

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Springs Series C. Table VI Calibration Tests.

Loads in Pounds	Overall Heights Compression Front	Back	Average	Release Front	Back	Average	Loads in Pounds	Overall Heights Compression Front	Back	Average	Release Front	Back	Average
Spring C-1.							Spring C-2						
Free	8.421	8.438	8.430	8.371	8.377	8 374	Free	8.591	8.535	8.563	8.541	8.458	8.500
500	8.099	8.098	8.099	7.892	7900	7896	500	8.216	8.216	8.216	8.015	8.020	8.018
1000	7.720	7.715	7.718	7.451	7.451	7.451	1000	7.811	7 802	7.807	7.600	7.594	7.597
1500	7.354	7.347	7.351	7.054	7.041	7048	1500	7.509	7.498	7.504	7.204	7.193	7.199
2000	6 967	6.953	6.960	6663	6.650	6657	2000	7.160	7.148	7 154	6.826	6.815	6 821
2500	6.614	6600	6 607	6.282	6.269	6.276	2500	6 761	6 749	6.755	6.458	6.444	6.451
3000	6.158	6.145	6 152	5.917	5903	5900	3000	6 392	6 380	6 386	6 085	6.073	6 079
3500	5.743	5730	5737	5.606	5594	5600	3500	5.963	5.951	5.957	5.755	5.743	5.749
							4000	5.529	5 508	5 514	5.430	5.419	5.425
Solid	5.141	5 134	5.138				Solid	5.244	5 236	5 240			
Spring C-3							Spring C-4						
Free	8.351	8274	8 313	8.315	8.209	8.262	Free	8.302	8.313	8.308	8.235	8.293	8.264
500	7.956	7.951	7.954	7.755	7.758	7757	500	7.938	7.951	7.945	7.761	7.777	7.769
1000	7.579	7.568	7.574	7 330	7 324	7 327	1000	7.580	7.582	7.581	7.349	7.352	7.351
1500	7.208	7.192	7.200	6.914	6902	6908	1500	7.200	7.195	7 198	6.912	6.907	6.910
2000	6.800	6785	6.793	6 520	6.507	6.514	2000	6.810	6 805	6 808	6 550	6 545	6.548
2500	6 372	6.357	6 365	6.126	6.115	6.121	2500	6.425	6.420	6.423	6.215	6.209	6 212
3000	5.928	5.915	5.922	5.721	5.708	5.715	3000	5.981	5.976	5.979	5.789	5.783	5.786
3500	5.441	5.428	5.435	5.342	5.331	5.337	3500	5.528	5.522	5.525	5.449	5.438	5.444
Solid	5.124	5.115	5.120				Solid	5.218	5.211	5.215			
Spring C-5.							Spring C-6.						
Free	8 423	8.443	8 433	8 386	8 391	8.389	Free	8.184	8.212	8.198	8,142	8.157	8.150
500	8 035	8 041	8.038	7.974	7.990	7 982	500	7.808	7.824	7.816	7711	7.732	7722
1000	7.726	7.726	7726	7.560	7.565	7.563	1000	7.483	7.490	7.487	7276	7283	7 280
1500	7.378	7.371	7.375	7.154	7 150	7.152	1500	7.113	7.113	7.113	6 876	6.876	6 876
2000	7 008	7 004	7006	6.757	6.755	6.756	2000	6758	6758	6.758	6 499		6.499
2500	6.607	6.602	6.605	6.377	6.374	6.376	2500	6.370		6.370	6 139		6.139
3000	6.233	6 230	6 232	6.028	6.026	6.027	3000	5.951		5 951	5.755		5.755
3500	5.829	5.826	5.828	5.664	5.662	5.663	3500	5.547	5.546	5 547	5.418	5.418	5.418
4000	5.406	5.402	5.404	5.338	5.335	5.337							
Solid	5.229	5.230	5.230				Solid	5.101	5.101	5 101			
Spring C-7.							Spring C-8						
Free	8.050	8 035	8.043	8 025	7.999	8 012	Free	7803	7765	7784	7.761	7741	7751
500	7698	7.712	7.705	7.607	7.625	7 616	500	7 459	7.421	7 440	7.390	7.354	7.372
1000	7357	7 366	7.362	7233	7.241	7.237	1000	7.113	7 063	7 088	7.004	6.954	6.979
1500	7008	7 010	7 009	6870	6 870	6.870	1500	6.779	6.722	6.751	6.653	6 595	6.624
2000	6 669	6.669	6.669	6.507		6.507	2000	6.442	6.385	6.414	6.311	6.254	6.283
2500	6.302	6 302	6.302	6.162	6.162	6.162	2500	6.093	6.037	6 065	5.951	5.893	5.922
3000	5.945	5.945	5.945	5.811		5 811	3000	5.735	5.678	5.707	5.643	5.595	5.619
3500	5.604	5.603	5.604	5.519	5.519	5 519	3500	5.380	5.323	5.352	5.329	5.270	5 300
Solid	5.260	5.262	5.261				Solid	5.215	5.155	5.185			

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Springs Series C. Table VI Page Two

Loads in Pounds	Overall Heights Compression Front	Back	Average	Release Front	Back	Average	Loads in Pounds	Overall Heights Compression Front	Back	Average	Release Front	Back	Average
Spring C-9							Spring C-10						
Free	7 911	7.894	7.903	7.877	7.861	7.869	Free	7.925	7.975	7.950	7.903	7.931	7 917
500	7.595	7.562	7.579	7.515	7.488	7.502	500	7.626	7.600	7.613	7.530	7.504	7.517
1000	7.251	7.208	7.230	7.137	7.097	7.117	1000	7.282	7 246	7 264	7.168	7.134	7.151
1500	6.910	6.857	6.884	6.763	6.712	6.738	1500	6.956	6.914	6.935	6.822	6.780	6.801
2000	6.556	6.503	6.530	6.397	6.343	6.370	2000	6.605	6.560	6.583	6.452	6 408	6.430
2500	6.196	6.143	6.170	6.062	6.009	6.036	2500	6.252	6.207	6.230	6.114	6.070	6.092
3000	5.837	5.784	5.811	5.750	5.695	5.723	3000	5.882	5.837	5.860	5.768	5.723	5.746
3500	5 469	5 416	5.443	5.426	5.374	5.400	3500	5.506	5.463	5.485	5.460	5 416	5 438
Solid	5.337	5.287	5.312				Solid	5.320	5.279	5.300			
Spring C-11							Spring C-12						
Free	7.537	7.436	7.487	7.501	7.411	7 456	Free	7.656	7.569	7 613	7.625	7.537	7.581
500	7.209	7.187	7.198	7.088	7.068	7.078	500	7.281	7.268	7.275	7.259	7.240	7.250
1000	6.859	6.826	6.843	6.736	6.705	6.721	1000	6.969	6.945	6.957	6.907	6.877	6.892
1500	6.489	6.448	6.469	6.386	6.343	6.365	1500	6.652	6.617	6 635	6.587	6.562	6.575
2000	6.194	6.150	6 172	5.997	5.952	5.975	2000	6.325	6.294	6.310	6.232	6.205	6.219
2500	5.866	5.822	5.844	5.732	5.685	5.709	2500	5.999	5.967	5.983	5.905	5.872	5.889
3000	5 531	5.488	5.510	5 417	5.372	5 395	3000	5.651	5.614	5.633	5.590	5.565	5.578
3500	5.200	5.157	5.179	5.155	5.113	5.134	3500	5.374	5.354	5.364	5.330	5.304	5.317
Solid	5.123	5.102	5.113				Solid	5.216	5.200	5.208			
Spring C-13							Spring C-14						
Free	7.534	7.550	7.542	7.517	7.516	7.517	Free	7.684	7.665	7.675	7.652	7.634	7.643
500	7.219	7.207	7.213	7.155	7.140	7.148	500	7.338	7.318	7 328	7.303	7.287	7.295
1000	6.871	6.843	6.857	6.781	6.757	6.769	1000	7.059	7.024	7.042	6.960	6.927	6.944
1500	6.522	6.485	6.504	6.426	6 388	6 407	1500	6 710	6.666	6.688	6.602	6 565	6.584
2000	6.197	6 163	6.180	6.085	6.058	6.072	2000	6.391	6 336	6.364	6.260	6.221	6.241
2500	5.841	5.800	5.821	5.750	5.723	5.737	2500	6 027	5.987	6.007	5.942	5.903	5.923
3000	5.492	5.456	5.474	5.423	5.385	5.404	3000	5.710	5.674	5.692	5.637	5.596	5.617
3500	5.204	5.166	5.185	5.201	5.175	5.188	3500	5.401	5.360	5.381	5.368	5.326	5.347
Solid	5.185	5.152	5.169				Solid	5.319	5.287	5.303			

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Springs Series C. — Table VI — Page Three.

Deflection Readings on Four Active Coils.

Loads in Pounds	Spring C-1 Compression	Spring C-1 Release	Spring C-5 Compression	Spring C-5 Release	Spring C-10 Compression	Spring C-10 Release	Spring C-13 Compression	Spring C-13 Release
Free	5.39	5.35	5.32	5.28	5.05	5.03	4.82	4.81
500	5.17	5.02	5.06	5.01	4.85	4.76	4.61	4.57
1000	4.93	4.72	4.88	4.72	4.61	4.53	4.38	4.31
1500	4.71	4.46	4.65	4.45	4.40	4.30	4.13	4.07
2000	4.44	4.20	4.42	4.21	4.17	4.06	3.92	3.84
2500	4.23	3.95	4.16	3.95	3.93	3.84	3.67	3.60
3000	3.93	3.71	3.90	3.73	3.70	3.61	3.42	3.37
3500	3.66	3.50	3.64	3.50	3.43	3.40	3.20	3.21
Solid	3.10		3.18		3.26		3.17	

Spring Measurements, and Development of Constants.

Spring No.	C-1	C-2	C-3	C-4	C-5	C-6	C-7
Size of Bar (1)	.744"	.750"	.735"	.738"	.745"	.745"	.744"
Outside Diameter (2)	4.581"	4.594"	4.592"	4.550"	4.571"	4.574"	4.554"
Mean Helix Diameter (3)	3.778"	3.785"	3.802"	3.756"	3.769"	3.775"	3.758"
Spring Index (4)	5.078	5.047	5.173	5.089	5.059	5.067	5.051
Rover-Wahl Correction Factor	1.3050	1.3072	1.2986	1.3042	1.3064	1.3058	1.3069
M.F.S. per 1000-lb Load — Conventional Formula	23362	22845	24381	23799	23211	23248	23239
M.F.S. per 1000-lb Load — with Rover-Wahl Corr.	30487	29863	31661	31039	30323	30357	30371
Capacity Solid - lbs.	4220	4315	3820	3840	4200	4045	3980
M.F.S. Solid — Conventional Formula	98590	98580	93135	91390	97490	94040	92490
M.F.S. Solid — with Rover Wahl Correction	128660	128860	120945	119190	127360	122790	120880

Spring No.	C-8	C-9	C-10	C-11	C-12	C-13	C-14
Size of Bar (1)	.748"	.749"	.746"	.746"	.750"	.736"	.747"
Outside Diameter (2)	4.553"	4.552"	4.552"	4.550"	4.546"	4.518"	4.534"
Mean Helix Diameter (3)	3.757"	3.753"	3.754"	3.760"	3.750"	3.734"	3.740"
Spring Index (4)	5.026	5.011	5.036	5.040	5.000	5.076	5.007
Rover-Wahl Correction Factor	1.3087	1.3097	1.3079	1.3077	1.3105	1.3052	1.3100
M.F.S. per 1000-lb Load — Conventional Formula	22904	22744	23074	23061	22634	23850	22650
M.F.S. per 1000-lb Load — with Rover-Wahl Corr.	29974	29788	30178	30157	29662	31129	29672
Capacity Solid - lbs.	3730	3680	3740	3585	3650		
M.F.S. Solid — Conventional Formula	85430	83700	86300	82670	82610		
M.F.S. Solid — with Rover-Wahl Correction	111800	109620	112870	108110	108270		

(1) Average of six measurements.
(2) Average of six measurements, not corrected for pitch angle.
(3) Corrected for pitch angle (perpendicular to helical axis)
(4) Ratio ~ Mean Helix Diameter/Size of Bar

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Springs Series — Table VII — Tests at Wright Field

Spring No	Free Height	Solid Height	Total Deflection	Ratio Deflection/Solid Hgt	Set in Inches — Static Loading — Compression No 1	Compression No 2	Compression No 3	Fatigue Loading — No of Cycles 6000	No of Cycles 10000	Total Set	Load in lbs used in Fatigue Test	Stress Rover-Wahl Formula	Number of Cycles to Failure
C-1	830	520	310	60	08	03	.02	.06	.00	.19	3650	111000	73600
C-2	838	525	3.13	.60	.07	03	06	.02	00	18	2750	82000	384400
C-3	824	510	3.14	62	.07	.02	.02	.06	.00	.17	2840	89800	249300
C-4	818	525	293	.55	04	.03	03	-.05(1)	.00	.05	2260	70000	931200
C-5	835	525	310	.59	05	.02	.01	.02	.00	.10	3300	99800	115300
C-6	8.10	510	3.00	.59	.04	.03	.01	.01	.00	.09	3140	95000	147200
C-7	800	525	275	52	.02	00	.02	.06	.00	10	3830	116000	138000
C-8	772	520	252	48	02	01	01	.06	.00	.10	3700	110000	148200
C-9	783	530	253	48	03	02	00	.03	02	10	3385	100000	209800
C-10	789	530	259	49	02	.02	01	06	03	.14	3496	105000	127400
C-11	7.45	5.10	235	46	06	.00	00	.02	.01	.09	2668	80000	486200
C-12	758	520	238	46	02	01	.02	-.02(1)	00	.03	2208	65000	No failure 1507500

(1) Growth

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Springs Series D. Table VIII Calibration Tests.

Loads in Pounds	Overall Heights: Compression Front	Compression Back	Compression Average	Release Front	Release Back	Release Average	Loads in Pounds	Overall Heights: Compression Front	Compression Back	Compression Average	Release Front	Release Back	Release Average
Spring D-1							Spring D-3.						
Free	9.575	9.637	9.606	9.526	9.536	9.531	Free	9.367	9.347	9.357	9.302	9.292	9.297
500	9.290	9.296	9.293	9.116	9.127	9.122	500	9.066	9.077	9.072	8.936	8.950	8.943
1000	8.939	8.935	8.937	8.744	8.737	8.741	1000	8.751	8.752	8.752	8.639	8.645	8.642
1500	8.600	8.587	8.594										
2000	8.252	8.241	8.247	8.009	8.000	8.005	2000	8.032	8.026	8.029	7.960	7.955	7.958
2500	7.913	7.901	7.907										
3000	7.568	7.557	7.563	7.300	7.291	7.296	3000	7.400	7.394	7.397	7.154	7.149	7.152
3500	7.223	7.213	7.218										
4000	6.881	6.870	6.876	6.616	6.608	6.612	4000	6.722	6.717	6.720	6.544	6.539	6.542
4500	6.520	6.509	6.515										
5000	6.165	6.155	6.160	5.931	5.920	5.926	5000	6.029	6.023	6.026	5.843	5.838	5.841
5500	5.800	5.789	5.795										
6000	5.424	5.413	5.419	5.333	5.342	5.338	6000	5.371	5.365	5.368	5.325	5.320	5.323
Solid	5.222	5.214	5.218				Solid	5.141	5.146	5.144			
Spring D-4							Spring D-5						
Free	8.946	8.951	8.949	8.933	8.932	8.933	Free	8.980	8.954	8.967	8.957	8.917	8.937
500	8.605	8.615	8.610	8.501	8.520	8.511	500	8.640	8.657	8.649	8.573	8.593	8.583
1000	8.272	8.280	8.276	8.149	8.158	8.154	1000	8.304	8.306	8.305	8.230	8.234	8.232
2000	7.587	7.581	7.584	7.465	7.461	7.463	2000	7.612	7.609	7.611	7.528	7.524	7.526
3000	6.989	6.985	6.987	6.830	6.829	6.830	3000	7.006	7.003	7.005	6.876	6.873	6.875
4000	6.315	6.311	6.313	6.186	6.184	6.185	4000	6.329	6.327	6.328	6.219	6.218	6.219
5000	5.666	5.664	5.665	5.560	5.560	5.560	5000	5.688	5.685	5.687	5.626	5.624	5.625
Solid	5.188	5.206	5.197				Solid	5.214	5.215	5.215			
Spring D-6							Spring D-7						
Free	9.068	9.127	9.098	8.860	9.102	8.881	Free	7.762	7.773	7.768	7.760	7.769	7.765
500	8.801	8.814	8.808	8.617	8.642	8.630	500	7.440	7.460	7.450	7.457	7.477	7.467
1000	8.432	8.435	8.434	8.285	8.293	8.289	1000	7.156	7.168	7.162	7.139	7.149	7.144
2000	7.781	7.774	7.778	7.598	7.595	7.597	1500	6.802	6.805	6.804	6.824	6.824	6.824
3000	7.126	7.120	7.123	6.919	6.917	6.918	2000	6.494	6.494	6.494	6.506	6.506	6.506
4000	6.497	6.491	6.494	6.282	6.280	6.281	2500	6.195	6.195	6.195	6.210	6.210	6.210
5000	5.824	5.818	5.821	5.663	5.661	5.662	3000	5.928	5.930	5.929	5.905	5.905	5.905
							3500	5.642	5.642	5.642	5.635	5.635	5.635
Solid	5.230	5.250	5.240				Solid	5.322	5.329	5.325			
Spring D-8							Spring D-9						
Free	7.690	7.723	7.707	7.687	7.723	7.705	Free	7.822	7.861	7.842	7.820	7.859	7.840
500	7.372	7.391	7.382	7.380	7.401	7.391	500	7.518	7.542	7.530	7.512	7.535	7.524
1000	7.025	7.033	7.029	7.071	7.077	7.074	1000	7.199	7.212	7.206	7.195	7.209	7.202
1500	6.730	6.728	6.729	6.744	6.741	6.743	1500	6.865	6.869	6.867	6.880	6.883	6.882
2000	6.454	6.451	6.453	6.418	6.416	6.417	2000	6.562	6.562	6.562	6.573	6.573	6.573
2500	6.144	6.141	6.143	6.142	6.139	6.141	2500	6.257	6.258	6.258	6.250	6.251	6.251
3000	5.808	5.805	5.807	5.818	5.816	5.817	3000	5.930	5.931	5.931	5.951	5.951	5.951
3500	5.524	5.521	5.523	5.519	5.516	5.518	3500	5.638	5.638	5.638	5.654	5.654	5.654
Solid	5.254	5.257	5.256				Solid	5.313	5.319	5.316			

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Springs Series "D" Table VIII Page 2.

Deflection Readings on Four Active Coils.

	Spring D-1 Compression	Spring D-1 Release	Spring D-3 Compression	Spring D-3 Release	Spring D-9 Compression	Spring D-9 Release
Free	6.29	6.25	6.13	6.09	4.82	4.82
500	6.11	5.96	5.94	5.86	4.66	4.65
1000	5.86	5.69	5.74	5.63	4.46	4.40
1500	5.63				4.24	4.21
2000	5.41	5.23	5.28	5.15	4.03	4.03
2500	5.21				3.84	3.80
3000	4.96	4.76	4.88	4.62	3.63	3.60
3500	4.75				3.41	3.38
4000	4.53	4.32	4.42	4.25		
4500	4.30					
5000	4.03	3.82	3.95	3.78		
5500	3.76					
6000	3.47	3.39	3.41	3.32		
Solid	3.25		3.12		3.12	

Spring Measurements, and Development of Constants.

Spring No.	D-1	D-3	D-4	D-5	D-6	D-7	D-8	D-9
Size of Bar (1)	.753"	.749"	.752"	.752"	.750"	.751"	.752"	.750"
Outside Diameter (2)	4.619"	4.611"	4.596"	4.594"	4.586"	4.557"	4.563"	4.556"
Mean Helix Diameter (3)	3.787"	3.793"	3.778"	3.776"	3.771"	3.758"	3.764"	3.757"
Spring Index (4)	5.033	5.064	5.024	5.021	5.028	5.004	5.005	5.009
Rover-Wahl Correction Factor	1.3082	1.3059	1.3088	1.3090	1.3085	1.3102	1.3102	1.3099
MFS per 1000-lb Load: Conventional Formula	22632	22986	22621	22609	22761	22591	22537	22676
MFS per 1000-lb Load: with Rover-Wahl Correct'n	29607	30017	29606	29595	29783	29599	29528	29703
Capacity Solid ~ lbs.	6270	6270	5725	5710	5930	4085	3970	4055
M.F.S. Solid: Conventional Formula	141900	144120	129505	129100	134970	92280	89470	91950
M.F.S. Solid: with Rover-Wahl Correction	185635	188210	169490	168990	176610	120910	117230	120445

(1) Average of six measurements

(2) Average of six measurements, not corrected for pitch angle.

(3) Corrected for pitch angle (perpendicular to helical axis)

(4) Ratio ~ Mean Helix Diameter / Size of Bar

Progress Report No Three — Section V

Springs Series D. — Table IX — Tests at Wright Field

Spring No	Free Height	Solid Height	Total Deflection	Ratio Deflection / Solid Hgt	Set in Inches: Static Loading, Compression No. 1.	2.	3.	Fatigue Loading, No. of Cycles 6000	10000	Total Set	Load in lbs. used in Fatigue Test	Stress-Rover Wahl Formula	Number Cycles to Failure
D-1	955	522			Not	tested							
D-2	Rejected		on	preli	minary	tests							
D-3	928	5.8	410	.791	.02	.04	.00	.00	.00	.06	2008	60000	1750000 No failure
D-4	890	528	362	.686	.03	.01	.01	.00	.00	.05	2380	70000	438300
D-5	892	524	308	737	.04	01	.00	00	.00	.05	4108	120000	34700
D-6	907	532	375	705	.02	.01	01	00	.00	.04	3717	110000	81100
D-7	775	534	241	452	01	.00	.01	.00	.00	.02	3064	90000	120800
D-8	768	522	246	472	02	.00	.00	.0c	00	02	3413	100000	154900
D-9	784	534	250	468	02	00	.00	00	00	.02	2713	80000	189200

Brinell Hardness on Machined Flats – 425 to 436

TABLE 10 PHYSICAL PROPERTIES OF THE FOUR GROUPS OF STEELS

Group	A and B	C	D
Physical properties in tension:			
Proportional limit, lb per sq in.	118,000	105,000	90,000
Yield point, lb per sq in.	142,300	134,500	150,000
Ultimate tensile strength, lb per sq in.	206,200	205,190	205,040
Elongation per cent	7	7	12
Modulus of elasticity, lb per sq in.	29,100,000	30,600,000	32,500,000
Physical properties in torsion:			
Proportional limit, lb per sq in.	82,500	60,000	108,000
Yield point, lb per sq in.	119,000	94,500	129,000
Modulus of rupture, lb per sq in.	153,100	164,180	175,000
Modulus of rigidity, lb per sq in.	11,800,000	12,300,000	11,800,000
Endurance:			
Rotating-beam, lb per sq in.	100,000	93,000	102,000
With 60-deg V notch, lb per sq in.	64,000	80,000	70,000
Torsional, lb per sq in.	110,000	90,000	....

NOTE: Groups *A* and *B*—Plain carbon basic open-hearth steel.
Group *C*—Plain carbon electric-furnace steel.
Group *D*—Silicon-vanadium steel.

TABLE 11 COMPARISON OF RESULTS OF TESTS ON JOHNSON'S AND THE A.S.M.E. SUBCOMMITTEE'S SPRINGS

	For 1,000,000 cycles, lb per sq in.	Probable endurance limit, lb per sq in.
Results of tests on Johnson's springs of:		
Plain carbon electric-furnace steel	98,000	93,000
Chrome-vanadium steel	82,000	77,000
Silicon-manganese steel	75,000	70,000
Plain carbon acid open-hearth steel	73,000	68,000
Results of tests on the Committee's springs of:		
Plain carbon basic open-hearth steel	...	72,730
Silicon-vanadium steel	...	62,820
Plain carbon electric-furnace steel	...	58,170

Air Resistance of Railroad Equipment

By A. I. LIPETZ,[1] SCHENECTADY, N. Y.

The author reviews the fundamentals of railroad aerodynamics and the work done by previous experimenters in this field. He then presents the results of wind-tunnel tests conducted jointly by the American Locomotive, American Car & Foundry, and J. G. Brill companies at New York University, but devotes the greater part of the paper to drag tests, the object of which was the evaluation of the resistance of air to the motion of locomotives, cars, and trains. After determining the air-resistance formulas for the models used in the tests, the author expands the coefficients of the formulas to apply to full-scale equipment, and further simplifies the formulas for application to the prototypes of the models. He compares air-resistance of full-scale equipment as obtained from the formulas derived from test results and the simplified formulas, presenting such data in tabular as well as graphical form. The author concludes his paper with a discussion of air-resistance tests of full-size equipment and savings in power effected by streamlining.

Introduction

FORMULAS for the evaluation of resistance of locomotives and cars appeared first in the middle of the last century. They consisted of two members and generally could be expressed

$$R = A + CV^2$$

where R is the total resistance, V is the speed of the vehicle or train, and A and C are certain constants. In the first known formula, that of Clark (1),[2] which was enjoying general recognition for about half a century, R was expressed in pounds per weight of train in long tons, V was speed in miles per hour, $A = 6$, and $C = 1/240$.

The Erfurt (2) formula, which was very popular in Germany and continental Europe, also had two members with $A = 2.4$ and $C = 1/1300$, if V were expressed in kilometers per hour and R in kilograms per metric ton. Later, when train speeds began to go up, Frank, and later Strahl, in Germany (3), Barbier (4) in France, Sanzin (5) in Austria, and Aspinall (6) in England suggested three-member formulas of the general form

$$R = A + BV + CV^2$$

where A, B, and C are constants depending upon the consist and the configuration of the trains. These formulas are now in use in Europe.

In America the Baldwin Locomotive Works (7) and the American Locomotive Company (8); formulas have been in use for a long time. Lately, also the Davis (9) three-member formula has come into vogue.

In all these cases the member with V^2 represented partly or wholly the air resistance. It was thus known in the sixties of the last century that the air resistance is proportional to V^2. Eiffel, of Eiffel Tower fame, made very exhaustive tests, dropping bodies of different shapes from high altitudes, and found that the force resisting the movement of bodies in air is proportional to the square of the velocity. This further confirmed the theoretical considerations that the air resistance of railway vehicles[3] should be represented by a member with V^2.

In recent times the wind tunnels, the existence of which was brought about by aerodynamical studies necessary for developments in the field of aeronautics, offered an opportunity for studying air resistance of vehicles, both for use on railways and highways. The first attempt at building a wind tunnel for these purposes, prior to any wind tunnel built for aeronautical studies, was made by the famous American locomotive experimenter W. F. M. Goss, who built one in 1896 at Purdue University for graduate students. The results of the tests were later reported by Goss (11, 12, 13).

The Purdue wind tunnel was small; it had a rectangular section of 20 × 20 in. and was 60 ft in length. The models were rather crude; they were made as rectangular smooth painted boxes, representing the bodies of freight cars, to a scale of $^1/_{32}$ in., the sizes being $12^1/_{16}$ in. long, $3^3/_8$ in. wide, and $4^1/_2$ in. high. The models were kept still while the air was blown through the tunnel by a powerful Sturtevant blower, giving air velocities up to 150 fps (over 100 mph). The technique of the tests was very remarkable for those days. The drag was accurately measured by a system of levers and the proper distribution of air velocities throughout the tunnel section was thoroughly studied and registered. A great many factors were already known to Goss. He found, for instance, that the total resistance offered by the air to a train consists of a quite important frontal resist-

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until December 10, 1937, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

[3] The only statement to the effect that air resistance varies with V in the first power ever made can be found in F. W. Carter's book (10) in which he gives a series of curves based on tests made in 1905 and 1906 with cars on the New York Central Railroad. This may be true, within certain limits, for low speeds.

ance of the first car, of a suction resistance of the last car, and of varying resistances on intermediate cars, depending upon their location and their length. Dr. Goss was also perfectly aware of the skin-friction resistance offered by the sides and tops of the car, but did not come to study the ground effect and did not pay sufficient attention to the importance of the protuberance of various parts of locomotives and cars. Although the formulas which he established for air resistance of locomotives and cars were very remarkable for those days, they are now obsolete and should not be applied to our present-day railway equipment.

That the wind-tunnel practice with small models was permissible for drawing conclusions regarding air resistance for large-size vehicles was proved later, long after Goss made his tests. The effect of the Reynolds number and the effect it has on questions of similarity and relativity regarding tests with models was developed only when aerodynamical problems began to be studied in wind tunnels. A very good and complete exposition of these questions was presented by Tietjens (14). In this paper Tietjens showed that the coefficients of fluid resistance of bodies of geometrically similar shape are equal, if the Reynolds number in each case is the same. For high velocities the coefficients remain practically constant, and if the models are not too small their test results are quite applicable to full-size vehicles, because the variation in Reynolds' numbers permits the law of similarity to be applied with a very small error.

The questions of railroad aerodynamics have been further elucidated by Klemin (15). In discussing the "scale effect," which is another expression for the error tolerated when model-test results are extrapolated to full-size locomotives and cars, Klemin states that: "The question of scale effect in wind tunnel tests of trains does not appear to have anything like the importance sometimes imputed to it." In his opinion, the error does not exceed 2 per cent for the best possible streamlined trains.

Thus, in 1932, when the first German high-speed, lightweight streamlined train was built and put in experimental service between Berlin and Hamburg, and when this example was followed by the Union Pacific and Burlington Railroads in this country by ordering their lightweight aluminum and stainless-steel trains, engineering science in Europe and America was ready for the development of these new ideas. Railroads became interested in high-speed, lightweight trains, and inquiries for streamlined locomotives began to appear. The research department of the American Locomotive Company, which had then already been in existence for more than a decade, had been following for some time the developments in high-speed vehicles in other than railroad fields in different countries.

W. C. Dickerman, president, and J. B. Ennis, vice-president, of the American Locomotive Company, conceived then the idea of making tests in order to establish values of possible economy in power due to streamlining, and the research department of that company started to investigate the available tunnels in this country with a view to making such tests. A number of tunnels were visited, and taking into consideration the size, equipment, and geographical location of the tunnels with respect to the main and engineering offices of the American Locomotive Company, the Daniel Guggenheim School of Aeronautics of New York University, with its aerodynamic wind tunnel, was chosen for the tests.

About the same time, the American Car & Foundry Company became interested in making tests with streamlined cars, and the J. G. Brill Company, manufacturers of rail cars, were also anxious to study streamlining power cars for Diesel-electric trains. It was, therefore, suggested that these two companies join the American Locomotive Company and make these tests jointly with both locomotive and car models.

Early in 1934 a committee of representatives of all three companies was formed for the study of the problem and for carrying out the tests. Messrs. V. R. Willoughby, general mechanical engineer of the American Car & Foundry Company, and G. O. Guernsey, then chief engineer of the J. G. Brill Company, were made members of the committee, and the author of this paper was appointed chairman. George DeBell, engineer of the American Car & Foundry Company, also served on the committee and took active part in the execution of the tests. The actual carrying out of the tests was left to the personnel of the Guggenheim School of Aeronautics, New York University, under the supervision of Prof. Alexander Klemin and the committee.

Fundamentals of Railroad Aerodynamics

On the basis of the aerodynamical theory and from experimental investigations it is known that a train running at a certain speed through still air encounters first, front-end resistance of air, sometimes called head resistance or form resistance, due to air pressure on the frontal area of the first vehicle or locomotive; second, friction resistance on the sides and roofs of the cars of the train, called skin-friction resistance; third, a suction effect on the last car due to the vacuum which is caused by the train pulling through the air; and fourth, the resistance caused by the air eddies and turbulence underneath the car called, in general, boundary-layer effects. All these resistance forces combined together offer what is called air or wind resistance, and sometimes air drag. If wind is encountered, an increase or decrease of resistance results, depending upon the direction of the wind.

It is also known that the first, third, and fourth components of air resistance are proportional to the square of the speed and can be represented by a formula

$$R_1 = a_1AV^2 \qquad [1]$$

where A is the frontal area, sq ft of the locomotive or first car; V is the speed, mph; and a_1 is the engineering drag coefficient of resistance. For skin-friction resistance a similar formula can be established

$$R_2 = a_2LV^2 \qquad [2]$$

where L is the length of the train, V is the speed as before, and a_2 is another engineering coefficient.

Strictly speaking, formula [2] is approximate. According to Klemin, skin-friction resistance of a well-streamlined train can be represented by

$$R_{sf} = a_{sf}L^{0.85}V^{1.85} \qquad [2a]$$

where a_{sf} is still another engineering coefficient.

In some cases, of which we shall see examples later, formulas [1] and [2] can be combined into an approximate one

$$R = (aA + bL^n)V^2 \qquad [3]$$

or even

$$R = KV^2 = aAV^2 \qquad [4]$$

where A is the frontal area of the vehicle, sq ft; V is the speed mph; and a is an engineer's coefficient. Klemin (15) and Pawlowski (16) give a more scientific justification for the foregoing formulas, but later simplify them by using what they call "engineer's coefficient." The simplification permits the engineer to eliminate the rather involved conceptions of such factors as kinematic viscosity and Reynolds' numbers, which are of no importance for the present and can be omitted by the practical engineer. They will be discussed later in so far as it is necessary for the subject of the paper.

The experimental railroad aerodynamics is based on two main principles:

1 The first principle is that of dynamic relativity. Ac-

cording to this, the aerodynamic phenomena do not change if the relative motion of air and bodies remains the same; in other words, it is immaterial whether the body is moving in still air, or whether the body is stationary and the air and surroundings are moving in the opposite direction with the same relative speed to the body. Dr. Goss in his wind-tunnel tests accepted that as an axiom. Afterward Eiffel made tests with bodies by dropping them from the Eiffel Tower and also by measuring the resistance of the still bodies in a wind tunnel. Very good agreement was obtained for both cases (17).

2 The second principle refers to dynamic similarity. This means that air resistance of bodies of geometrically similar shape is identical, if the Reynolds number in each case is the same. However, if the Reynolds number is not the same, the phenomena may still be approximately the same for a certain region of the values of the Reynolds numbers beyond the critical (14).

On the basis of these two principles it is possible to establish air resistance of different bodies by testing them in wind tunnels. For railway vehicles the strict application of the principles permits for testing the use of: (*a*) Immovable models of railway vehicles (locomotives, cars, trains); (*b*) air moving with a required speed in a direction opposite the movement of the vehicles; and (*c*) a plane endless belt representing the track and moving with a speed and direction of the air in the tunnel.

Wind-Tunnel Tests

Since Dr. Goss' tests were made (11), very little experimental work with models of railroad motive power and equipment had been carried out for a long time. In 1927 Prandtl made some tests in the Göttingen, Germany, wind tunnel with models of high-speed electric cars (18) and Tietjens, his former collaborator, made tests in 1930 in the Westinghouse Research Laboratory with models of short high-speed trains and interurban cars (19).

Prandtl found that, for two streamlined models of cars, the total air-resistance coefficients in formula [4] were $a = 0.00110$ for one model, and $a = 0.00105$ for another model. The models were made to $^1/_{25}$ scale and the cross-sectional area of the full-size car was 107.6 sq ft. Tietjens and Ripley found that for a streamlined electric locomotive of a cross-sectional area of 126 sq ft, the air-resistance coefficient was $a = 0.0009$, and that for each streamlined car this coefficient has to be increased by 0.00038. The coefficient for a nonstreamlined locomotive and the increase per nonstreamlined car should be 0.00204 and 0.00102, respectively.[4]

In 1932, the Borsig Locomotive Works, before starting the construction of their streamlined locomotives (class 005), which since have become famous, had the Berlin-Charlottenburg Technical Institute make wind-tunnel tests for them with locomotive and one- and two-car models. Two types of locomotives were tested, one with a tender and another of the tank type; a standard and streamlined locomotive of each type was used. The locomotives had a cross-sectional area of approximately 116.0 sq ft. The locomotive with the tender was 113 ft long while the tank locomotive was 62.5 ft long. It was found that for the nonstreamlined standard locomotive $a = 0.00264$; for the streamlined standard locomotive $a = 0.00114$; and for the streamlined tank locomotive $a = 0.00105$. Subsequently, Nordmann of Reichsbahn-Zentralamt published more accurate figures (20) which do not differ essentially from these figures.

Later, the Berlin Technical Institute continued the tests with models of cars (21) for themselves, and obtained very interesting results, among them that the effect of cars does not follow the law of proportionality and the straight line, but depends upon the location of the cars in the train, as previously found by Goss and other experimenters.

Nordmann also reports on tests made with models of the Flying Hamburger, the first streamlined lightweight, Diesel-electric train, placed in trial service in the fall of 1932 (20).

The Union Pacific Railroad which simultaneously was following similar practice, had tests made at New York University, but no reports have been published. Some results are quoted by Klemin (15) and resistance coefficients a fluctuating between 0.000887 and 0.001038 were found (22) which were close to the results of previous German tests. Likewise, tests were made in this country with models of streamlined trains which have been appearing since on many railroads. While it is known that tests were made at the wind tunnels of the Massachusetts Institute of Technology, the University of Michigan, Case School of Applied Science, and many other places, published reports are lacking.

Exhaustive tests with locomotive models were made in 1931 by the National Research Laboratories, Ottawa, Canada, for the Canadian National Railways. These tests are remarkable for the size of the model, which was to one-twelfth scale, probably the largest in recent tests, and for the importance of obtained results (23). They were also significant for the addition of smoke tests and the development of special smoke-stack streamlining and for the shaping of the top of the boiler with the intent of carrying away the smoke. The resistance coefficient a referred to the frontal area was, for a standard nonstreamlined locomotive, as high as 0.00268, while after streamlining and modifications the coefficient was reduced to 0.00174, which meant a reduction in resistance of 35 per cent due to streamlining.

Other test results are also given by Klemin in his article (15).

Quite recently tests made by the Advisory Committee for Scientific Research of the London, Midland & Scottish Railway were conducted with models in a 7-ft wind tunnel of the National Physical Laboratory on the joint behalf of the London, Midland & Scottish, the London & North Eastern, and the Southern Railways, and a report of the tests was made later by Johansen (24). These tests also cover a great deal of investigation on the influence of oblique winds (yaw tests). Reference to these tests will be made later.

Wind-Tunnel Tests of the American Locomotive, American Car & Foundry, and the J. G. Brill Companies

The tests referred to at the beginning of the paper were carried out in the 9-ft wind tunnel of the Daniel Guggenheim School of Aeronautics at New York University. This tunnel is of the closed-throat double-return type (25) with a working section originally 9 ft long which is straight and octagonal in section. The rear part is built as a cone, fairing into a circular section 14 ft in diameter. The front part of the cone was rebuilt for the tests into a straight 23-ft-long octagonal section, forming a continuation of the working part; thus, uniform air speed for the longest train of models was obtained. A longitudinal section through the tunnel is shown in Fig. 1. The 13-ft 8-in. eight-blade propeller is driven by an electric motor which permitted during the test, a maximum speed of 75 mph.

The air velocity was measured at the tests by a pitot static tube which is connected to a liquid manometer mounted on the balance platform and filled with colored alcohol. Knife-edge balances of the drag-and-lift type are installed in the tunnel for measuring the three forces and three moments along and around three ortogonal axes. Each balance is operated by a $^1/_{200}$-hp motor, controlled by electric contact points at the ends of the balance arms, permitting them to float around a position corresponding to the balanced load, which is then very accurately measured. Details are given in a description of the tunnel published by Teichmann and Ruffner (26).

[4] The foregoing figures are quoted from Klemin's article (15), p. 314.

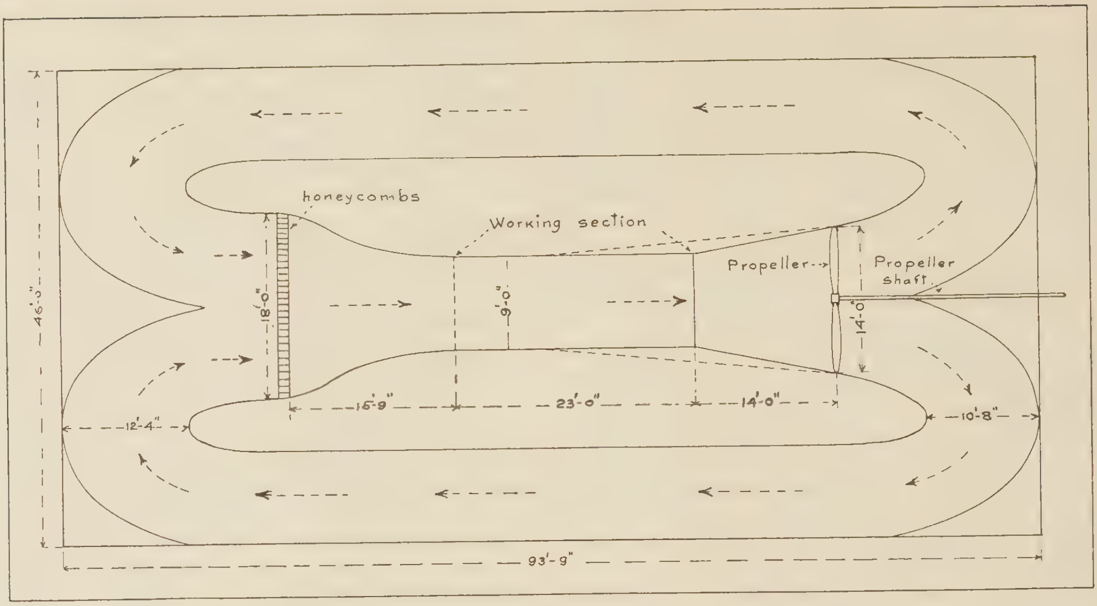

Fig. 1 Section Through the Wind Tunnel at New York University

SCALE OF MODEL	1/32 SIZE 3/8" = 1 FT																	1/16 SIZE 3/4" = 1 FT														1/32 SIZE 3/8" = 1 FT				
TEST NO.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36
STD. LOCO.																												X	X	X	X	X	X	X	X	
STREAM LINE LOCO.																X	X				X	X	X	X	X	X	X									X
POWER CAR	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X			X	X	X																
TAIL CAR	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X										X
NO. OF STD. COACHES																													1	2	3		1	2	3	
NO OF STREAM LINE COACHES	2	2	2	2	2	1		3	4		1	2	2	1				2	1																	
OPEN SIDE SILL SKIRTS			X	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O										O
CLOSED SIDE SILL SKIRTS	X	X																																		
STD TRUCK	X		O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O										O
FAIRED TRUCK (POWER CAR)		X																																		
TAIL FAIRING #1	X	X	X		O	O	O	O	O	O	O	O	?	?	?	O	O	O	O	O	O	O	O	O	O	O										O
" " #2				X																																
POWER CAR NOSE #1	X	X	X	X		O	O	O	O	O	O	O	O	O	O			O	O	O																
" " " #2					X																															
HELMET NOSE																X	X				X															
ROUND NOSE																						X		O	O	O	O									O
STRAIGHT NOSE																							X													
ROUND TOP																X					X	X	X		O	O	O									O
COWLED TOP																	X							X												
CLOSED SHROUD (LOCO. & TENDER)																X	X				X	X	X	X												
OPEN SHROUD " "																									X											O
SHORT SHROUD " "																										X	O									
SMOKE TEST																X	X																			
VELOCITY RUN INCLUDED (0° YAW)	X	X	X	X	X	X	X	X	X									X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
YAW TEST (0° TO MAX ANGLE)	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X															X	X	X	X	X
DRAG TEST																		X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	
TEST TAIL CAR-PRESENCE OF TRAIN										X	X	X	X	X	X	X	X																			
NOTE: X=TYPE TESTED O=BEST TYPE ?=WORST TYPE																																				

Fig. 2 Schedule of Tests Conducted Jointly by the American Locomotive, American Car & Foundry, and J. G. Brill Companies

Three types of models were used for the tests: First, in order to determine the best method of testing, block models which had been made to a 1/32 scale were used; these models were made without details, only to represent streamlined vehicles of the proper length and contour. Second, small models to a 1/32 scale were made for determining the influence of oblique winds; they were called "yaw models," as they permitted their convenient placing at yaw in the extended working part of the tunnel. Third, larger models were used for getting as reliable figures as possible for locomotive and car resistance in the available length of the train. While we started out with the desire of making the size of the models to a 1/12 scale, further consideration compelled the committee to reduce the size of the models to a 1/16 scale. These models were called "drag models."

The type of equipment was so selected and the sequence of tests was so arranged that every feature of streamlining could be tested one at a time, after the effect of other features had been found from previous tests, thus permitting the selection of the

best type for each individual feature while the tests were still in progress. The schedule of tests was so worked out that the number of tests was minimum for a maximum number of combinations. The schedule is shown in Fig. 2.

For instance, for the streamlined locomotive a round-top boiler and a cowled-top boiler in which the cowl covered the dome, sandbox, and cab turret, were tested. As to the front part, a helmet nose, round nose, and straight nose were experimented with. In order to obtain all possible combinations of tops and noses, the models were parted as described later in the paper. For obtaining the straight nose, special pieces in the lower part of the front were provided to enable exchanging them without disturbing the rest of the models. Shrouding of the locomotive wheels and mechanism was foreseen, as well as openings for oiling the running mechanism in a full-size locomotive. Therefore, the following shrouds were provided:

(a) A shroud which covered the sides between locomotive cylinders and cab; called "long (or closed) shroud."

(b) A shroud which left open the sides from the cylinders to the cab; called "short shroud."

(c) A shroud with openings for oiling; called "open shroud."

Modifications were made for the cars in the lower part of the body by covering it up with skirts (closed) or simply leaving it open. The trucks were equipped with side aprons on the power car and trailing cars (called faired trucks), or built of the standard type. On the streamlined locomotives the trucks on the tender were covered by the continuation of the locomotive shroud, except on the locomotive with the short shroud as shown in Fig. 10. Diaphragms between cars for small-size models ($^1/_{32}$) were smooth, while in drag models ($^1/_{16}$) they were of two types—smooth and corrugated All drag models with the previously mentioned modifications were built to $^1/_{16}$ scale. For yaw models the same equipment was duplicated to $^1/_{32}$ scale. Drag tests were also made with this smaller equipment (at zero yaw), and for this, a standard 4-6-4 locomotive was built to $^1/_{32}$ scale n order to permit a comparison between the test results at the two different scales. Thus, the following types of equipment were studied:

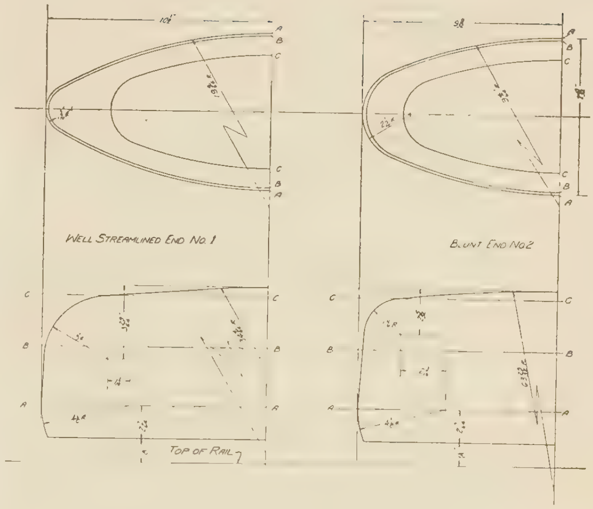

Fig. 5 Shape of Tail-Car Ends

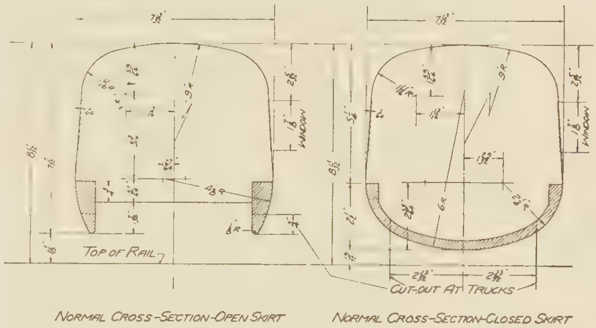

Fig. 6 Closed and Open Side-Sill Skirts

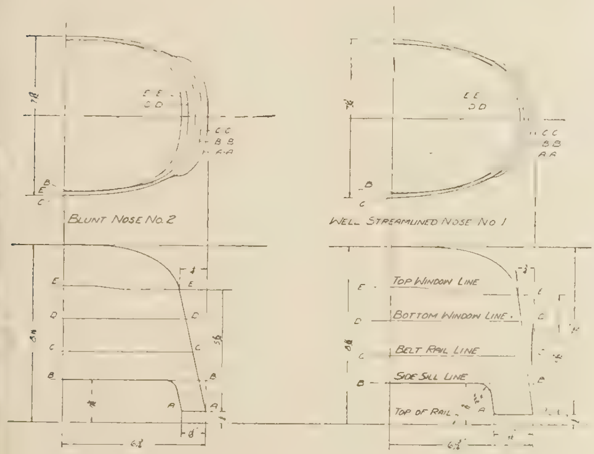

Fig. 3 Power-Car Noses

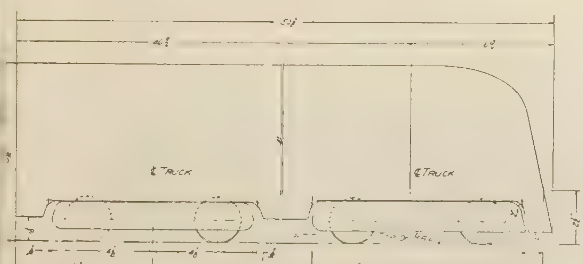

Fig. 4 Drag Model of Power Car With Faired Trucks

1 Streamlined power car with the two types of streamlined nose shown in Fig. 3, and with standard and faired trucks shown in Fig. 4. Models made to $^1/_{16}$ scale.

2 Tail car with two types of tail as shown in Fig. 5, and with standard or faired trucks. Models made to $^1/_{16}$ scale.

3 Intermediate streamlined cars with standard and faired trucks and also open and closed skirts as shown in Fig. 6. Models made to $^1/_{16}$ scale.

4 Streamlined 4-6-4 Hudson-type locomotive with three types of nose, i.e., helmet, round, and straight as shown in Figs. 7A, 7, and 12; two types of boilers, i.e., round top and cowled top as shown in Figs. 7, 9, and 10; and three types of shroud, i.e., long, short, and open (windows) as shown in Fig. 7. Models made to $^1/_{16}$ scale.

5 Standard 4-6-4 Hudson-type locomotive shown in Fig. 8 made to $^1/_{16}$ scale.

6 Standard cars which were tested with the standard locomot ve. Models made to $^1/_{16}$ scale.

7 Streamlined 4-4-4 locomotive with round-top and cowled-top boiler, also shown in Fig. 7. Model made to $^1/_{32}$ scale.

8 Standard 4-6-4 locomotive built to $^1/_{32}$ scale, also shown in Fig. 7.

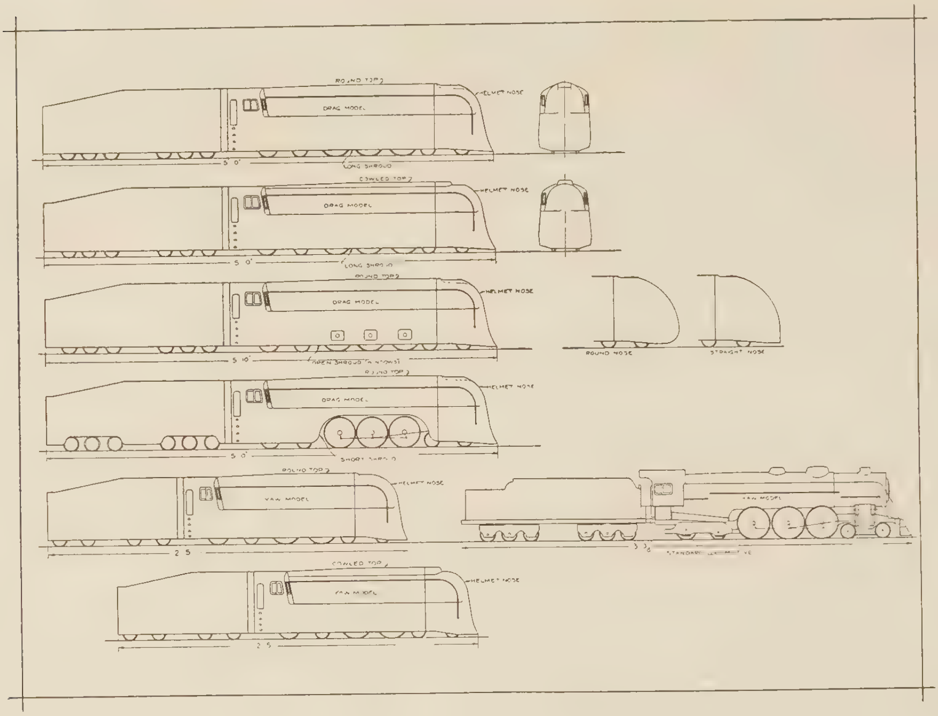

Fig. 7 Types of Locomotive Models Tested

The types of equipment and the difference between them are also illustrated in Figs. 9 to 12, inclusive.

Preliminary Tests

Different methods for testing railroad equipment are employed in wind-tunnel tests. The most popular are the ground representation by a fixed board in the direction of the wind, and the mirror, also called reflection method. Neither of the two is absolutely correct. The first has the disadvantage that the principle of dynamic relativity previously referred to under "Fundamentals of Railroad Aerodynamics" is not fulfilled, as it does not duplicate exactly the actual conditions. When a train is moving on rails with a speed V in relation to the ground, the ground and air are still and the relative velocity between the train on one hand, and the air and ground on the other hand, is V, while in the wind tunnel with a fixed ground board, the train and board are still and the relative velocity between the model and the air is V, but not between the model and the ground; it is only zero. Thus, the conditions are only partly reversed, and have not the same relation.

The mirror method seems to have a greater theoretical justification. It consists of mounting two identical models in the wind tunnel, with their surfaces placed in such relative position that the distance between wheels of the two models represents twice the ground clearance in the scale of the model. It is assumed that since the flow about the two models should be identical, there exists half-way between the undersurfaces of the model, a flow which lies entirely in a horizontal plane and is located at the point where the ground is to be represented. This is evident from Fig. 13 which is taken from Klemin's paper (27). Fig. 13 shows only that portion of Klemin's figure which illustrates the theoretical flow past two identical cylinders. If a true streamlined flow existed over the two models, this might be so, but the flow is not streamlined, as eddies which represent unstable and oscillatory flow conditions are liable to form. If the eddies should happen on both models at the same instant, the mirror method would still be theoretically correct, but, in fact, these eddies on two different models are not coincident in time and therefore not strictly symmetrical. Under these conditions, the eddies may cross the theoretical boundary and the ground condition is no longer simulated correctly.

A theoretically correct method is the method of a moving endless belt when the belt itself represents the ground relative to the model. If the upper surface of a belt is moving with a linear

Fig. 8 Standard Nonstreamlined Locomotive Model

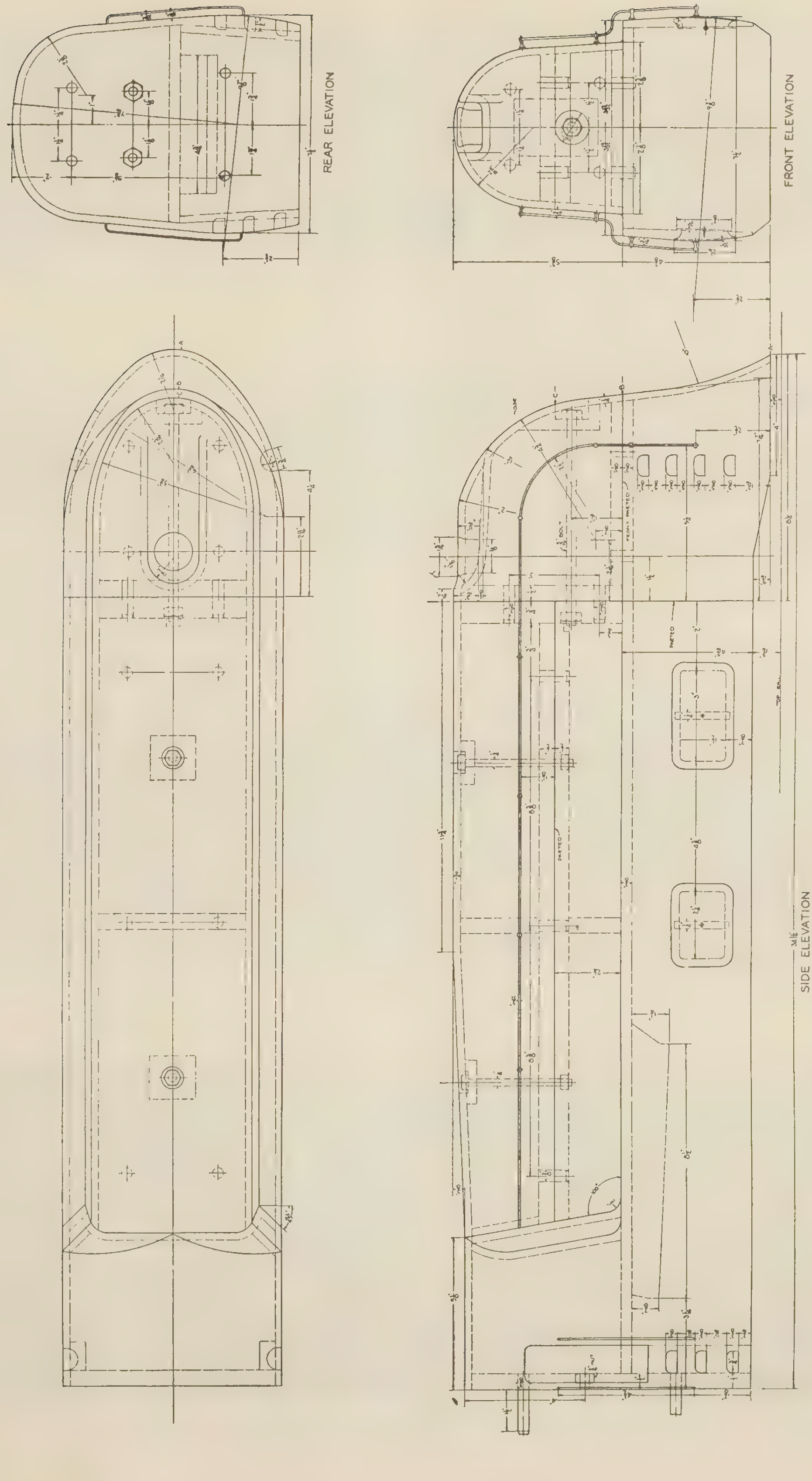

FIG. 7A MODEL OF STREAMLINED LOCOMOTIVE WITH HELMET NOSE, LUNG (CLOSED) SHROUD AND A ROUND-TOP BOILER

Fig. 9 Streamlined Locomotive Model With Cowled Boiler Top

Fig. 10 Streamlined Locomotive Model With Short Shroud

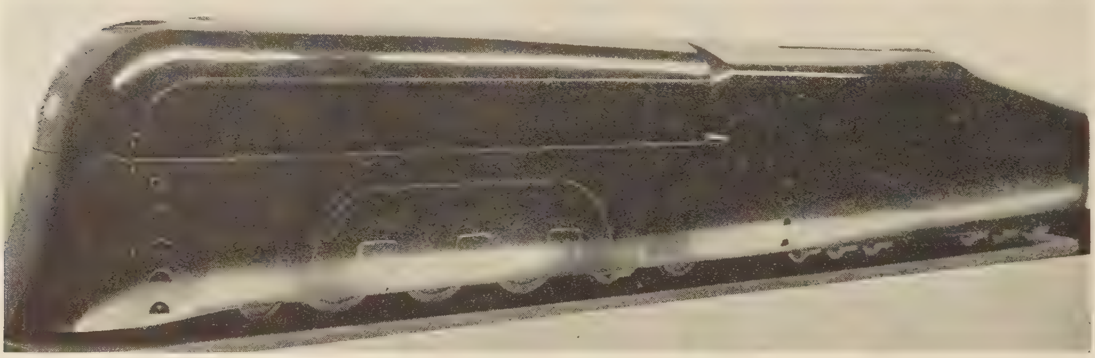

Fig. 11 Streamlined Locomotive Model With Detachable Sides and Windows

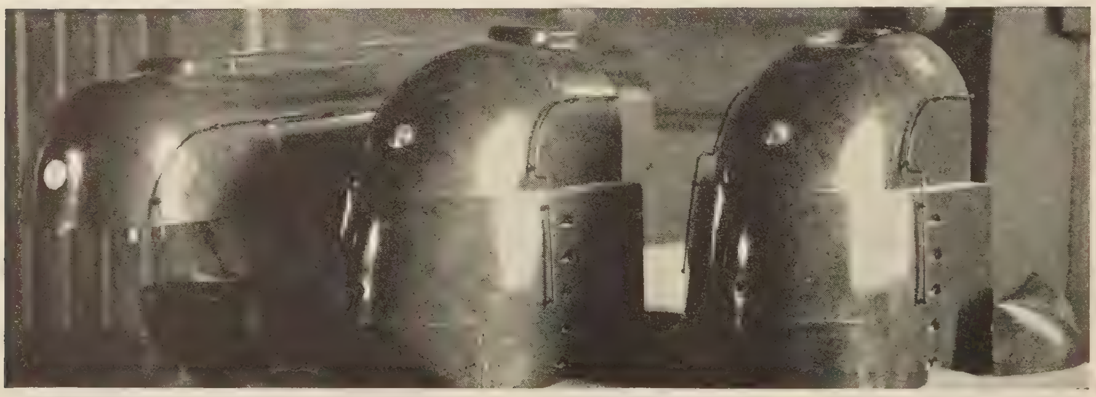

Fig. 12 Interchangeable Noses for Streamlined Models

speed V equal to the speed of the air, the relative velocity between the belt and the air is zero, while the relative velocity between the model and both the belt and air is V. Thus, the conditions are reversed as compared with the actual conditions but relatively identical with them. Such a method was suggested by Eiffel some twenty years ago, but tests made afterward failed to make the moving belt function properly, free from vibration, or shifting of position (28).

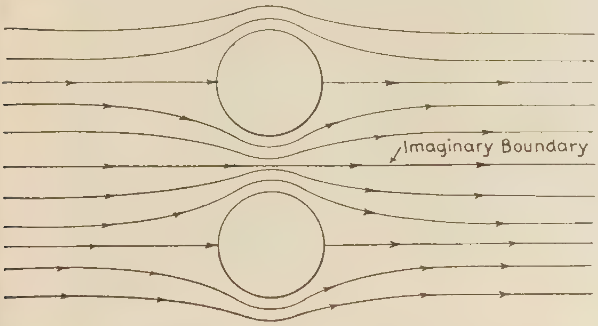

FIG. 13 THEORETICAL FLOW IN A MIRROR TEST WITH TWO CYLINDERS

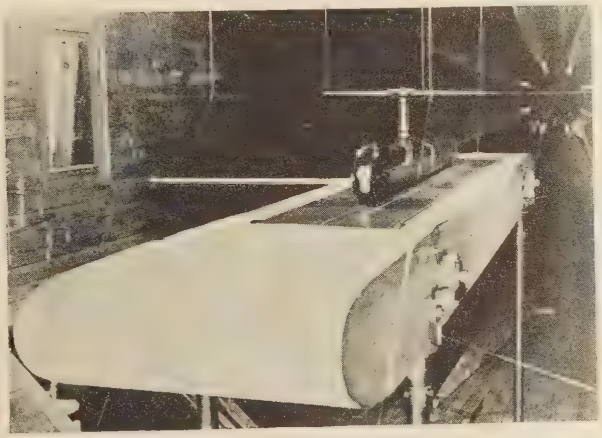

FIG. 14 LOCOMOTIVE MODEL AND TAIL CAR ON ENDLESS BELT IN WIND TUNNEL

Klemin, however, desired to try it again and therefore established in the Guggenheim aeronautical laboratory a moving belt 12 in. wide and 80 in. between centers of the two 7-in. rollers driven by two electric motors. By controlling the rotary speed of the rollers, the speed could be changed from 30 to 90 mph. Fig. 14 shows the installation of a power car and tail car on the moving belt in the tunnel, with the suction propeller in the background.

In addition to this test, three others were tried: (*a*) The first was with a fixed ground board of two different lengths, 40 in. and 120 in. × 60 in. wide, and at three different ground clearances of $^3/_{16}$, $^5/_8$, and $1^1/_8$ in. between the flats of the wheels and the ground board; (*b*) the second employed the mirror method. (*c*) the third employed a free suspension method. All three were made with a block model of a power car with a tail car. The results are shown in Fig. 15 by straight lines on logarithmic paper. The tangents of the straight lines represent the exponents of the drag values versus speed. The straight line for the belt test shows the conditions with the belt running, the straight line being directed through the points (triangles) as close as possible. Circles represent points when the belt was stopped. It can be seen that the circles fit the same straight line as well as the triangles do. Thus, there was practically no difference between the results under these two conditions in so far as it could be represented by straight lines on logarithmic paper. The difference between the line with the belt stopped and the board at the same clearance is probably due to the difference in the width of the board and the belt. The line of the belt drag values lies between those of the large board at a clearance distance of $^3/_{16}$ in. and $^5/_8$ in. between the flats of the wheels of the model and the ground board, the

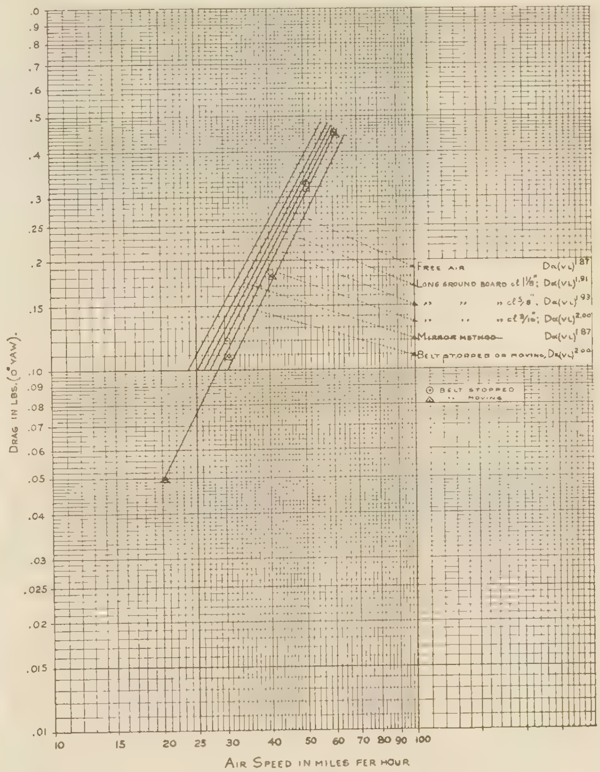

FIG. 15 TEST RESULTS OBTAINED BY DIFFERENT TESTING METHODS

lower limit being the ground clearance for a 6-in-high rail at a $^1/_{32}$ scale.

Thus, it was assumed that the drag measures of the model were unaffected (within experimental limits) by the velocity of the ground relative to the model and it was recommended that further tests be run with a fixed ground in the normal position (29). As this method of testing was the simplest and would give sufficiently accurate results, it has been adopted for the tests. A ground board 7 ft wide and 23 ft long was used, as some trains exceeded 20 ft in length. After the installation of the ground board the velocities were again calibrated and the pressure gradient in the tunnel verified.

YAW AND DRAG TESTS

The tests were carried out during the summer months of 1934 and 36 tests, as shown in the schedule in Fig. 2, were conducted. Forces and moments along and around three ortogonal axes, with models as previously stated, were measured at various speeds and angles of yaw (including 0 deg yaw), except those tests which were made for drag only. The results were compiled in a report (30) presented to the three sponsors of the tests (American Locomotive Company, American Car & Foundry Company, and the J. G. Brill Company) and approved by the Committee.

TABLE 1 RESULTS OF DRAG-MODEL TESTS

Test number	Equipment	Skirt	Power Truck	Trailing truck	Diaphragm	Tail fairing	Nose	Boiler	Shroud	Scale	Kp	Kf	K	a	A in sq ft	Error of (Kv^2) in % at 30 mph	Error of (Kv^2) in % at 120 mph
18	P+T+2SC	Open	Faired	Standard	Smooth	•2	•2	—	—	1/16	.1116	0774	.1503	.001691	88.89	-4.9	+ 63
18-A	″	″	″	″	Cowled	″	″	—	—	″	.1149	.0774	.1536	.001728	88.89	-4.8	+.62
19	P+T+1SC	″	″	″	Smooth	″	″	—	—	″	.0882	.0614	.1189	.001337	88.89	-4.9	+.64
20	P+T	″	″	″	″	″	″	—	—	″	.0744	.0437	.0962	.001082	88.89	-4.4	+.51
21	SL+T	″	—	″	″	″	Helmet	Round Top	Closed	″	.1126	.0535	.1393	.001010	137.98	-3.6	+.58
22	″	″	—	″	″	″	Round	″	″	″	.1152	.0535	.1419	.001029	137.98	-3.6	+.50
23	″	″	—	″	″	″	Straight	″	″	″	.1147	.0535	.1414	.001025	137.98	-3.6	+.50
24	″	″	—	″	″	″	Helmet	Cowled Top	″	″	.1161	.0535	.1428	.001035	137.98	-3.5	+.57
25	″	″	—	″	″	″	″	Round Top	Open	″	.1156	.0535	.1423	.001032	137.98	-3.5	+.57
26	″	″	—	″	″	″	″	″	Short	″	.1308	.0535	.1575	.001142	137.98	-3.3	+.39
27	SL	—	—	—	—	—	″	″	Closed	″	.1094	.0363	.1275	.000924	137.98	-2.8	+.31
28	NL	—	—	—	—	—	—	—	—	″	.2400	.0421	.2611	.001906	136 96	-1.7	+ 21
29	NL+1NC	—	—	—	—	—	—	—	—	″	.3103	.0636	.3420	002498	136.96	-1.8	+.20
30	NL+2NC	—	—	—	—	—	—	—	—	″	.3709	.0830	.4124	.003011	136 96	-1.9	+.27
31	NL+3NC	—	—	—	—	—	—	—	—	″	.3975	.1023	.4486	.003276	136.96	-2.2	+.16
32	NL	—	—	—	—	—	—	—	—	1/32	.2337	.0421	.2547	.001859	136.96	-1.7	+.19
33	NL+1NC	—	—	—	—	—	—	—	—	″	.3025	.0636	.3343	.002441	136.96	-1.8	+ 23
34	NL+2NC	—	—	—	—	—	—	—	—	″	.3518	.0830	.3933	.002871	136.96	-2.1	+.26
35	NL+3NC	—	—	—	—	—	—	—	—	″	.4082	.1023	.4593	.003353	136.96	-2 2	+.27

Notation used: P - power car; T - tail car; SC - streamlined car; SL - streamlined locomotive; NL - non-streamlined (standard) locomotive; NC - non-streamlined (standard) car.

• Errors are in relation to formula $(K_pV^2+K_fV^{1.85})$.

TABLE 2 RESULTS OF TEST NO. 21 (30)

Test No. 21
Streamline locomotive
Tail car
Open side-sill skirts
Trailing car std. trucks
Tail fairing long CT No. 2
Smooth diaphragm

1/16-Size drag model test
Helmet nose
Round top
Closed shroud (locomotive and tender)
Length = 10.34 ft
Side area = 18.56 sq ft

Indicated velocity, mph	Calibrated velocity, mph	Observed drag, lb	ΔD_p, lb	Net drag, lb	D_{sf}, lb	D_p, lb	$\frac{D_p}{(V/16)^2}$
30	29.6	0.655	0.086	0.569	0.169	0.400	0.1169
40	39.8	1.170	0.140	1.030	0.291	0.739	0.1192
50	50.0	1.790	0.235	1.555	0.444	1.111	0.1134
55	55.1	2.160	0.293	1.867	0.530	1.337	0.1126
60	60.3	2.545	0.340	2.205	0.628	1.577	0.1110
65	65.8	3.000	0.399	2.601	0.738	1.863	0.1101
70	71.4	3.565	0.489	3.076	0.860	2.216	0.1110
75	77.1	3.995	0.544	3.451	0.978	2.473	0.1063
						Total	0.9005
						Avg	0.1126

Full-scale drag = $0.1126\ V^2 + 0.0535\ V^{1.85}$

TABLE 3 RESULTS OF TEST NO. 22 (30)

Test No. 22
Streamline locomotive
Tail car
Open side-sill skirts
Trailing car std. trucks
Tail fairing long CT No. 2

1/16-Size drag-model test
Smooth diaphragm
Round nose
Round top
Closed shroud (locomotive and tender)
Length = 10.34 ft
Side area = 18.56 sq ft

Indicated velocity, mph	Calibrated velocity, mph	Observed drag, lb	ΔD_p, lb	Net drag, lb	D_{sf}, lb	D_p, lb	$\frac{D_p}{(V/16)^2}$
30	29.6	0.640	0.086	0.554	0.169	0.385	0.1122
40	39.8	1.155	0.140	1.015	0.291	0.724	0.1169
50	50.0	1.835	0.235	1.600	0.444	1.156	0.1181
55	55.1	2.205	0.293	1.912	0.530	1.382	0.1167
60	60.3	2.595	0.340	2.255	0.628	1.627	0.1143
65	65.8	3.045	0.399	2.646	0.738	1.908	0.1126
70	71.4	3.675	0.489	3.186	0.860	2.326	0.1169
75	77.1	4.165	0.544	3.621	0.978	2.643	0.1140
						Total	0.9217
						Avg	0.1152

Full-scale drag = $0.1152\ V^2 + 0.0535\ V^{1.85}$

TABLE 4 RESULTS OF TEST NO. 18 (30)

Test No. 18
Power car
Tail car
Two streamline coaches
Open side-sill skirts
Power-car faired truck
Trailing car std. trucks
Smooth diaphragms

1/16 Size drag-model test
Tail fairing, long CT No. 2
Power-car nose No. 2
Length = 17.24 ft
Side area = 28.71 sq ft

Indicated velocity, mph	Calibrated velocity, mph	Observed drag, lb	ΔD_p, lb	Net drag, lb	D_{sf}, lb	D_p, lb	$\frac{D_p}{(V/16)^2}$
30	29.6	0.740	0.140	0.600	0.241	0.359	0.1050
40	39.8	1.335	0.213	1.122	0.418	0.704	0.1135
50	50.0	2.115	0.331	1.784	0.635	1.149	0.1177
55	55.1	2.535	0.408	2.127	0.767	1.360	0.1150
60	60.3	2.990	0.480	2.510	0.902	1.608	0.1131
65	65.8	3.465	0.594	2.871	1.060	1.811	0.1072
70	71.4	4.215	0.713	3.502	1.230	2.272	0.1142
75	77.1	4.750	0.841	3.909	1.425	2.484	0.1069
						Avg	0.1116

Full-scale drag = $0.1116\ V^2 + 0.0774\ V^{1.85}$

TABLE 5 RESULTS OF TEST NO. 19 (30)

Test No. 19
Power car
Tail car
One streamline coach
Open side-sill skirts
Power-car faired truck
Trailing cars std. trucks
Smooth diaphragms

1/16 Size drag-model test
Tail fairing, long CT No. 2
Power-car nose No. 2
Length = 13.04 ft
Side area = 21.76 sq ft

Indicated velocity, mph	Calibrated velocity, mph	Observed drag, lb	ΔD_p, lb	Net drag, lb	D_{sf}, lb	D_p, lb	$\frac{D_p}{(V/16)^2}$
30	29.6	0.565	0.110	0.455	0.191	0.264	0.0772
40	39.8	1.030	0.157	0.873	0.332	0.541	0.0872
50	50.0	1.655	0.238	1.417	0.505	0.912	0.0934
55	55.1	1.990	0.301	1.689	0.606	1.083	0.0915
60	60.3	2.350	0.391	1.959	0.717	1.242	0.0874
65	65.8	2.740	0.458	2.282	0.805	1.477	0.0874
70	71.4	3.340	0.543	2.797	0.977	1.820	0.0915
75	77.1	3.840	0.615	3.225	1.130	2.095	0.0902
						Avg	0.0882

Full-scale drag = $0.0882\ V^2 + 0.0614\ V^{1.85}$

For lack of space, this paper is limited to only the drag tests, the object of which was the evaluation of the resistance of air to the motion of railroad motive power and equipment, that is, locomotives, cars, and trains. The ground wind should be mentioned here in so far as its effect in the direction of the train increases, or decreases the air resistance. DeBell (31) has already published a review of that part of the New York University report (30) which pertains to ground winds. He derived formulas for speed components and resultants and gave test curves for air drags with ground winds in relation to drags without wind, both for streamlined trains and trains composed of standard equipment. Interesting and corroborative results on ground winds have been reported by Johansen (24).

Of 36 tests, 20 were of the drag type. They are listed in Table 1. Fig. 16 represents a drag model of a streamlined locomotive with tender and tail car in the wind tunnel. Observations and

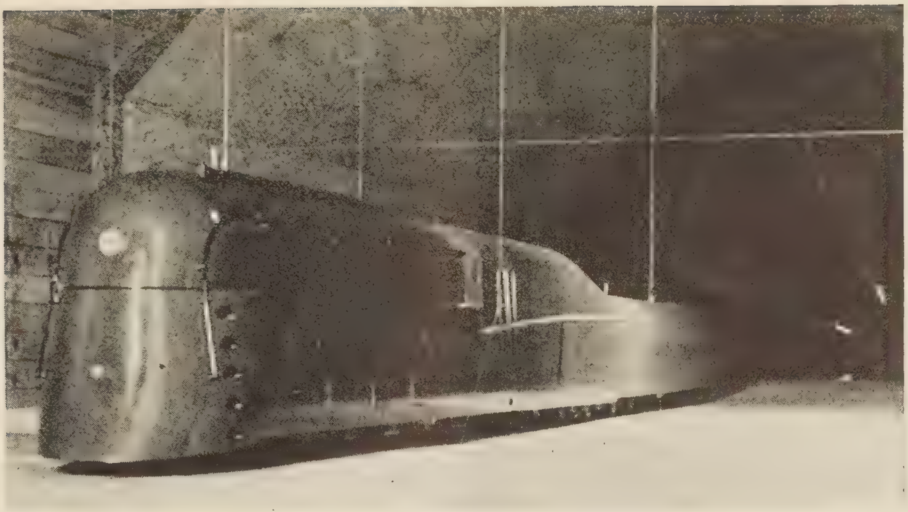

FIG. 16 DRAG MODEL OF LOCOMOTIVE WITH TENDER AND TAIL CAR IN THE WIND TUNNEL

computation results were put on separate sheets, one for each test. As an example, a copy of test sheet No. 21 is represented in Table 2. This pertains to a test with $^1/_{16}$-size drag model of a streamlined locomotive with a tail car. The locomotive had a helmet nose, a long (closed) shroud and a round-top boiler, shown in Figs. 7 and 7*A;* the car had open side-sill skirts, as shown in Fig. 6, and standard trucks. Between the tender and the car there was a smooth diaphragm. For purposes of comparison, test sheet No. 22 is shown in Table 3; it can be seen from the heading of Table 3 that the difference was only in the locomotive nose. Full-scale drag formulas as a result of the tests differed in this latter case only in the member with V^2, which had a coefficient 0.1152 instead of 0.1126 for the helmet nose, a difference of + 0.0026 for the round nose.

Table 4 represents a copy of sheet No. 18, which shows test results of a streamlined power car with nose No. 2 shown in Fig. 3, with tail-car end No. 2 shown in Fig. 5, open side-sill skirts shown in Fig. 6, faired trucks on the power car shown in Fig. 4, and two streamlined coaches with standard trucks. Table 5 shows the sheet of test No. 19 pertaining to the same equipment and details, but with only one coach instead of two. The difference between the values of these two formulas gives the saving in resistance of one intermediate streamlined coach.

Each of the test sheets, tables 2, 3, 4, and 5, has eight columns. The first gives the indicated velocity of air in miles per hour; the second column, the corrected velocity as established through calibration; third, the observed drag in pounds, as read from the electric scales; the fourth is the error ΔD_p due to pressure gradient in the tunnel, calculated for the length of the model given on the sheet; this has to be deducted from the observed drag. The difference between the preceding column and this latter column gives the net drag (fifth column). The sixth column represents the skin friction which is estimated for the wetted (side) area of the model indicated on the test sheet for the corresponding air velocity. The formulas for this calculation are given below. These figures are subtracted from the net drag to give, in the seventh column, the profile or pressure drag D_p in pounds. This is the force which is applied to the front part of the model due to the velocity of air, includes the boundary layer and the suction effects, and which, as we know, depends only upon the square of the speed. The last column represents the figures obtained by dividing the seventh column figures (D_p) by $(V/16)^2$, thus giving the coefficient of pressure drag for the first member of the next given formula.

ANALYSIS OF TEST RESULTS

The proper understanding of the test results might be facilitated if some preliminary calculations are considered and the Reynolds number is more closely scrutinized. By definition the Reynolds number

$$\text{R.N.} = \frac{\delta VL}{\mu} = \frac{VL}{\nu} \qquad [6]$$

where

δ = density of air = 0.002378 slugs per cu ft, slug being the English unit of mass; V = velocity, fps; L = length, ft, or any linear dimension of the model; μ = air viscosity = 0.000000373 lb sec per sq ft; and $\nu = \mu/\delta$ = 0.0001568 sq ft per sec. Thus

$$\text{R.N.} = 6378VL$$

if V as before is in fps. Also

$$\text{R.N.} = 9354VL$$

if V is in mph.

It is easy to see that the Reynolds number is nondimensional, i.e., not depending on the system of units; it is the same in the metric (centimeter-gram-second) or the English (foot-pound-second) system.

For models of our size, Reynolds' numbers fluctuate between 2,100,000 and 14,250,000, although sometimes in the wind tunnel, on account of turbulence, they might have been still higher. For our full-size equipment (locomotives, car, and trains) Reynolds' numbers vary between 88,200,000 and 304,000,000.

These figures may not mean much, but the interesting part of this is that the variation of the coefficient of friction between air and bodies is very slight, when Reynolds' numbers are as large as those just given. According to Tietjens (32) for a long cylinder, after a certain critical number, the coefficient remains almost constant as shown in Fig. 17. The same applies to coefficients of other bodies.

It is evident from the shape of the locomotive model in Fig. 7*A* that substantially a streamlined locomotive consists of very few, rather simple, curved surfaces, for which fortunately, we already have some coefficients from tests begun by

Eiffel and continued by other experimenters up to the present time. The top front part can be taken approximately as an upper half of a frontal surface of a half sphere. The resistance coefficient referred to the cross section of the sphere is 0.00045 according to Pawlowski (16) and 0.00051 according to the latest tests of Klemin, both in pounds per square foot. The lower part of the locomotive model can be considered to be a half-cylinder, with the axis perpendicular to the flow of air. As a cylinder, it will have a coefficient of resistance equal to 0.0009 (33). These

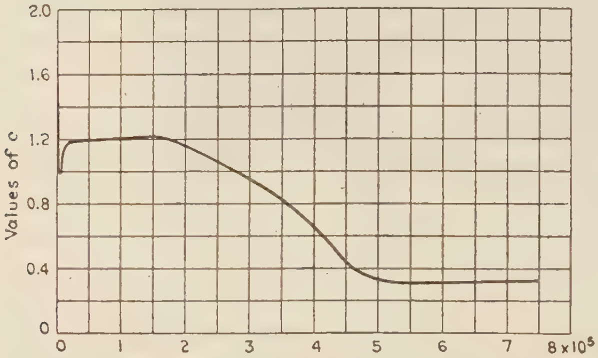

FIG. 17 COEFFICIENT OF AIR RESISTANCE FOR LONG CYLINDER AS A FUNCTION OF REYNOLDS' NUMBER

resistance coefficients are good for Reynolds' numbers over a million and, therefore, should at least be right for our models. The side of the locomotive, with the tender, in so far as it is shrouded, is a flat, presumably smooth, surface, placed in the plane of the wind. For this case the resistance is expressed by a formula

$$R_f = C_f \frac{\delta}{2} V^2 S \qquad [4a]$$

where R_f = skin-friction resistance, lb; C_f = the skin-friction drag coefficient, lb per sq ft; δ = density of air as previously stated, slugs per cu ft; V = speed in fps (1.4667 × mph); and S = wetted area on right and left side of the model, sq ft.

The skin-friction-drag coefficient can be figured approximately for large Reynolds' numbers (34)

$$C_f = 0.0375\ (\text{R.N.})^{-0.15} \qquad [5]$$

It is represented graphically in Fig. 18.

On the basis of these data, the air resistance of the helmet-nose model shown in Fig. 7*A* should be approximately:

1 Half sphere (center $2^{13}/_{16}$ in. from the top): Cross-sectional area and resistance for a speed of (1 mph)² are

$$\frac{1}{2}\pi \times (2^{13}/_{16})^2 = 12.425 \text{ sq in.}$$

$(1/144) \times 12.425 \times 0.00051 = 0.0000440$ lb per (1 mph)²

2 Two half-cylinders (from the center line of the sphere to the bottom): Cross-sectional area and resistance are

$$7^1/_2 \text{ (approx)} \times 4^7/_8 + 5^5/_8 \times 2^{15}/_{16} = 53.086 \text{ sq in.}$$

$(1/144) \times 53.086 \times 0.0009 = 0.0003318$ lb per (1 mph)²

3 Two flat surfaces of the rest of the locomotive and tender which surfaces (see also Fig. 7) have a length of 61.75 in. and a breadth (height) of 9.5625 in. in average: The wetted area (approximately) is

$$(1/144) \times 2 \times 61.75 \times 9.5625 = 8.2 \text{ sq ft}$$

and the skin-friction resistance is (the skin-friction coefficient for a Reynolds number of 4,100,000 being 0.00382).

$0.00382 \times (0.002378/2) \times (1.4667)^2 \times 8.2 = 0.0000801$ lb per (1 mph)²

Thus, the total air resistance of the model will be

$0.000044 + 0.0003318 + 0.0000801 = 0.0003758 + 0.0000801 = 0.0004559$ lb per (1 mph)²

The first member of the second sum represents the pressure-drag coefficient of the model k_p' less, however, the boundary-layer drag, while the last of the two members is the skin friction of the model k_f. The two together give a value for the total drag of the model, less the boundary-layer effect, or

$k_p' + k_f = 0.0003758 + 0.0000801 = 0.0004559$ lb per (1 mph)²

and per 1 sq ft of frontal area (12.425 + 53.086 = 65.511 sq in. = 0.455 sq ft)

$$a = \frac{0.0004559}{0.455} = 0.0010019 \text{ lb per sq ft per (1 mph)}^2$$

The air resistance of a well-streamlined locomotive with tender of a total length of 93.3 ft (length of model 5.83 ft) could thus be figured a priori, at least for a model, and found to be

$$R_a = aAV^2 \qquad [4]$$

where A = cross-sectional area of the model, sq ft; and a = 0.0010019, or 0.001, approximately.

This figure is remarkably close to the coefficients as found by Tietjens and Ripley for a streamlined electric locomotive alone

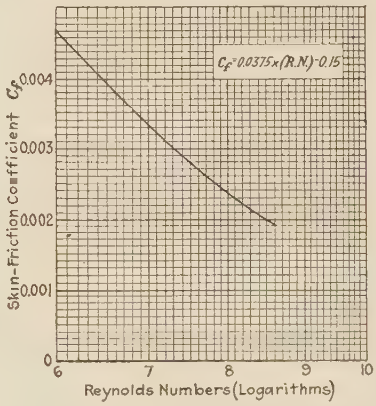

FIG. 18 SKIN-FRICTION COEFFICIENT

(0.0009), Prandtl (0.00105), and other experimenters, as given previously in this paper. It is also in good agreement with results of the tests discussed (Table 1, test No. 21, column *a*).

For large-size bodies (locomotive, cars, and trains), it would seem that we could simply increase the model drag figures in the ratio of the squares of the sizes, in this case multiplying them by 16² = 256. This would not be strictly correct because the skin-friction coefficient is not constant. It varies in accordance with formula [5]. Therefore, for the tests of New York University on behalf of the three companies, Klemin employed the following procedure: In the test sheets, examples of which were shown in Tables 2, 3, 4, and 5, the skin friction D_{sf} given in the sixth column, was figured for the model according to formulas [4*a*] and [5] in the same way as in the foregoing example. These figures were deducted from the net drag, given in the fifth column, and D_p, given in the seventh column, as well as $D_p/(V/16)^2$ the values in the eighth column were thus obtained. The average of these latter figures for speeds between 30 and 75 mph is the pressure-drag coefficient K_p for a full-size vehicle. This is because we know that the pressure drag is in proportion to the square of the linear dimensions and the square of speed, namely, $R_p = aAV^2$.

In regard to skin-friction drag of a full-size vehicle, D_{sf}, or simply R_f, this, according to formulas [4a], [5], and [6] combined, is

$$R_f = 0.0375 \frac{V^2}{2} S \left(\frac{VL}{\nu}\right)^{-0.15} \dots\dots\dots\dots\dots [7]$$

the notations being the same as in formulas [4a], [5], and [6]. For a model, the skin-friction drag is

$$r_f = 0.0375 \frac{v^2}{2} s \left(\frac{vl}{\nu}\right)^{-0.15} \dots\dots\dots\dots\dots [8]$$

the corresponding notations differing only by the size of the letters, that is, lower-case letters refer to the model. The ratio of the skin-friction value of a full-size vehicle to that of a model is

$$\frac{R_f}{r_f} = \left(\frac{V}{v}\right)^2 \frac{S}{s} \left(\frac{VL}{vl}\right)^{-0.15} = \left(\frac{V}{v}\right)^{1.85} \frac{S}{s} \left(\frac{L}{l}\right)^{-0.15}$$

It is evident that the ratio of wetted surfaces $\frac{S}{s} = \left(\frac{L}{l}\right)^2$

therefore

$$\frac{R_f}{r_f} = \left(\frac{V}{v}\right)^{1.85} \left(\frac{L}{l}\right)^2 \left(\frac{L}{l}\right)^{-0.15} = \left(\frac{V}{v}\right)^{1.85} \left(\frac{L}{l}\right)^{1.85}$$

The ratio of the linear dimensions (L/l) is the reciprocal of the scale size; in our case it is either $^1/_{16}$ or $^1/_{32}$. Denoting the ratio of the linear dimensions by $1/S_c$, we obtain

$$\frac{R_f}{r_f} = \left(\frac{V}{v}\right)^{1.85} \left(\frac{1}{S_c}\right)^{1.85}$$

whence

$$\frac{r_f}{v^{1.85} \times S_c^{1.85}} = \frac{R_f}{V^{1.85}} = K_f$$

If we denote the latter ratio by K_f, then

$$R_f = K_f \times V^{1.85}$$

from which we can determine R_f if we know the coefficient K_f, which in turn is calculated from the model skin friction

$$K_f = \frac{r_f}{(vS_c)^{1.85}}$$

where r_f = model drag friction, lb; v = model velocity, mph; S_c = scale of model ($^1/_{16}$ or $^1/_{32}$); r_f being obtained from formula [8].

The full-scale air resistance can then be established as

$$R_a = R_p + R_f$$

or

$$R_a = K_pV^2 + K_fV^{1.85} \dots\dots\dots\dots\dots\dots [9]$$

Values of K_p and K_f for all drag tests are given in Table 1. For instance, for a streamlined locomotive with tender and tail car (test No. 21), the formula is

$$R_a = 0.1126V^2 + 0.0535V^{1.85}$$

The coefficients 0.1126 and 0.0535 can be taken from Table 1 for this test.

This latter formula would give, for V = 100 mph, a total resistance of 1430 lb. As the cross section of the streamlined locomotive, the model of which was used for test No. 21, is 137.98 sq ft, the resistance coefficient a in formula [4] is

$$a = \frac{1430}{137.98 \times 10{,}000} = 0.001036$$

which is within 3.3 per cent of the a priori value of 0.0010019 found previously by simple calculations for the locomotive and tail car of the same length. As the boundary-layer effect was not included in the a priori air-resistance calculations, this also proves that this effect can be represented by the same coefficient a, if the cross-sectional area of the layer is included in the total area A. This may also explain the results of our moving-belt tests, when the boundary layer was brought into effect in one case and eliminated in the other, with no noticeable difference in results.

Full-Scale Formulas and Their Simplifications

The method of expanding coefficients from model tests to full-scale equipment, proposed by Klemin and explained previously is based on scientific grounds and has the advantage that coefficients K_p and K_f are constant, that is, they are the same for models and full-scale railroad equipment. In the report made by the Guggenheim School of Aeronautics (30), a formula of the two-member construction, such as formula [9], is given for each type of equipment as the result of each individual test; examples of the formulas are given in Tables 2, 3, 4, and 5. At this point the obligation of the School ceased. It was now our duty to make the best use of the report and formulas.

In the opinion of the author, one or two simplifications would be necessary, at least they were worth trying. He wished first to find out what is gained by going to the use of the two-member formula, in which one has the unusual exponent of 1.85, and whether the old one-member formula of the V^2 type would not do; in case it should, the author wanted to know what error this simplification would involve, if any.

In Fig. 19 is shown the curve resulting from test No. 18 made with a streamlined power car with tail car and two streamlined coaches. The formula recommended on the basis of test No. 18, results of which are given in Table 2, is

$$R_a = 0.1116V^2 + 0.0774V^{1.85} \dots\dots\dots\dots [10]$$

Crosses in Fig. 19 correspond to resistances for speeds at intervals of 10 mph from 10 to 120 mph, and it is evident that they can be very well represented by a parabolic (one-member) formula

$$R_a = KV^2 = 0.1503V^2 \dots\dots\dots\dots\dots [11]$$

The parabolic curve was found by the method of least squares. The differences in resistances obtained by the two formulas are as given in Table 6.

TABLE 6 COMPARISON OF AIR RESISTANCE FROM FORMULAS [10] AND [11]

Speed mph	R_a from formula [10], lb	R_a from formula [11], lb	Difference, per cent
10	16.6	15.0	−9.67
20	64.4	60.1	−6.63
30	142.3	135.3	−4.92
40	249.8	240.5	−3.72
50	386.6	375.8	−2.81
60	552.5	541.1	−2.07
70	747.4	736.5	−1.46
80	971.0	961.9	−0.93
90	1223.2	1217.4	−0.47
100	1503.9	1503.0	−0.06
110	1813.1	1818.6	+0.31
120	2150.6	2164.3	+0.64

This justifies the use of the one-member formula, because the difference is negligible; at low speeds the percentage difference may seem large, but actually it amounts to a few pounds. At high speeds (about 100 mph) it is less than a small fraction of one per cent. Besides, small speeds, below 50 mph, are now of little practical interest.

All two-member formulas derived from the tests were similarly analyzed and the results were the same. It was found that every two-member formula, of the type such as formula [9], can be replaced by a one-member formula of the type such as formula

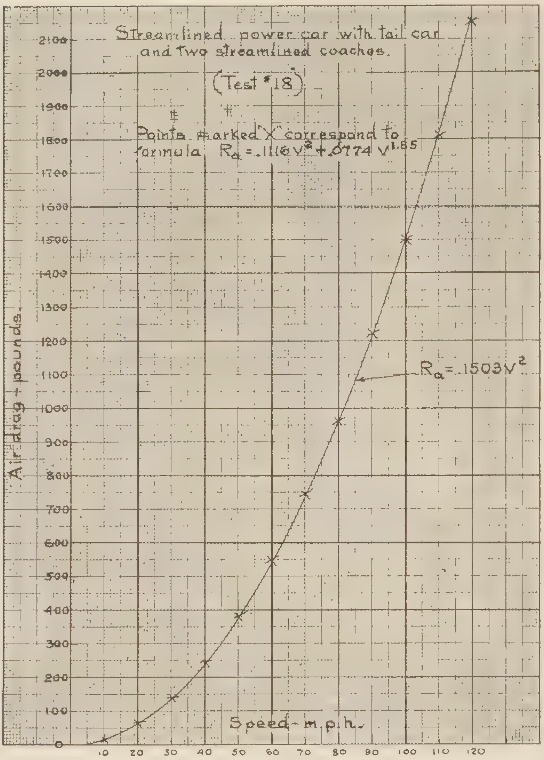

FIG. 19 COMPARISON OF RESISTANCE CURVES FROM TEST No. 18

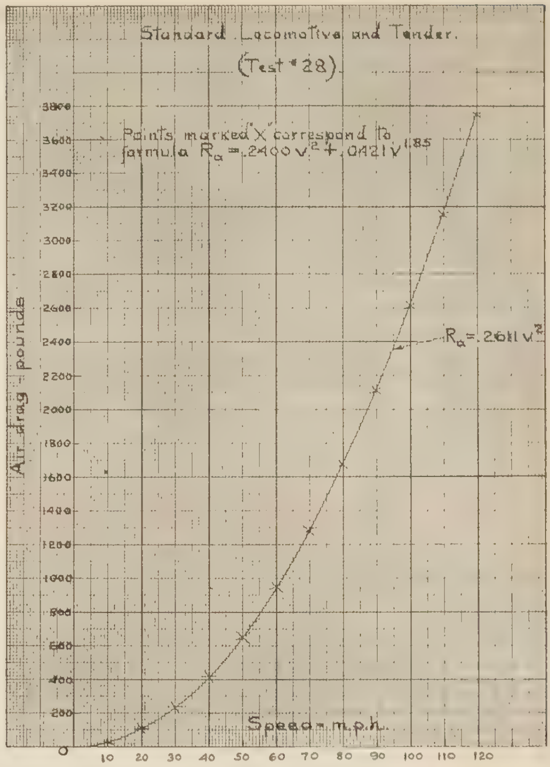

FIG. 20 COMPARISON OF RESISTANCE CURVES FROM TEST No. 28

[4]. This is true not only for streamlined equipment, but for nonstreamlined equipment as well. In Fig. 20 a comparison of curves for a standard locomotive and tender (test No. 28) is made. The formula resulting from the test and recommended in the New York University report (30) is

$$R_a = 0.2400V^2 + 0.0421V^{1.85} \qquad [12]$$

and is represented by the crosses for speeds between 10 and 120 mph. The curve shown in Fig. 20 is a one-member parabola

$$R_a = 0.2611V^2 \qquad [13]$$

The comparison of the two formulas is given in Table 7.

TABLE 7 COMPARISON OF AIR RESISTANCE FROM FORMULAS [12] AND [13]

Speed mph	R_a from formula [12], lb	R_a from formula [13], lb	Difference, per cent
10	27	26.1	−3.33
20	107	104.4	−2.39
30	239	234.9	−1.72
40	423	417.8	−1.23
50	659	652.7	−0.96
60	946	940.0	−0.63
70	1285	1274.4	−0.83
80	1676	1671.0	−0.30
90	2118	2114.9	−0.15
100	2611	2611.0	0
110	3156	3159.3	+0.10
120	3752	3759.8	+0.21

It seems that the air resistance of a nonstreamlined locomotive can be represented by a V^2 formula better than a streamlined locomotive. This can be explained by the fact that the nonstreamlined locomotive causes more turbulence, which follows the V^2 law, while the streamlined locomotive has more skin friction, wh ch follows the law with an exponent of 1.85. In either case the V^2 formula can be used for all practical purposes.

One remark would not be amiss here in this connection. It should not be thought that because of the admissibility of the V^2 law we could, for establishing the full-scale formulas, expand the coefficient from the model tests simply by increasing their values in the inverse ratio of the square of the scale, as this is being done by some experimenters. It can be easily seen by taking the net drag values of test No. 18 as given in Table 4 and multiplying them by $(16)^2 = 256$, that the figures thus obtained will be greater than the values of R_a in Table 6 for corresponding speeds. This is because that portion of the air resistance which is due to skin friction is then multiplied by 256, while it should be multiplied only by $(16)^{1.85} = 169$. By using the two-member formula with two different exponents (2 and 1.85) the scale effect was properly corrected, after which the simplification through the one-member formula was entirely permissible, as this gave practically identical numerical values.

Thus, all two-member formulas of the type

$$R_a = K_pV^2 + K_fV^{1.85} \qquad [4]$$

derived from the New York University tests (30) have been converted into one-member V^2 formulas of the form

$$R_a = KV^2 = aAV^2 \qquad [4]$$

Coefficients K_p, K_f, K, a, and A for each formula are given in Table 1. The maximum errors of this conversion at 30 and 120 mph are also shown in Table 1 for each test.

After the admissibility of the V^2 formula for all practical purposes has been established, the author asked Mr. DeBell, who was doing the mathematical computations for the committee, to try to simplify the formulas not only with respect to speed, but also to length, as the author was very much tempted by the straight-law hypothesis of Tietjens and Ripley (32) regarding the length of trains, both standard and streamlined, and would like,

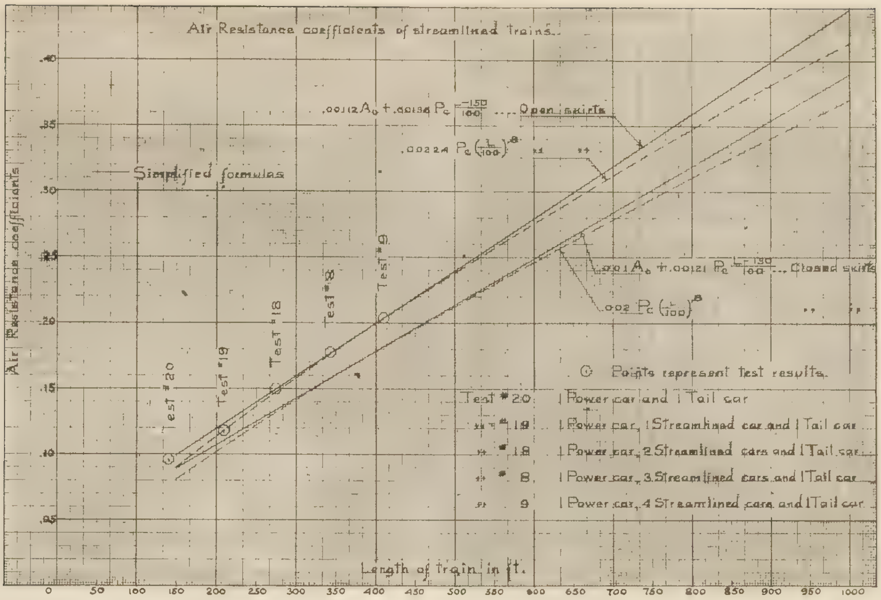

FIG. 21 AIR-RESISTANCE COEFFICIENTS OF STREAMLINED TRAINS

if possible, to follow it. It did not seem feasible, however, to discard the exponential functions in the member containing length of the trains, and, therefore, formulas on the V^2 basis, but with the exponent in the expression of length, were published by the author of this paper and Mr. DeBell in a joint article (35).

For power-car trains in still air, the following formulas were established:

For cars with open skirts (18 in. from skirt to top of rail) shown in Fig. 6

$$R_a = 0.00224P_c\left[\frac{L}{100}\right]^{0.8} V^2 + \Sigma KV^2 \qquad [14]$$

while for cars with closed skirts (completely under car), also shown in Fig. 6, a similar formula was recommended

$$R_a = 0.0020P_c\left[\frac{L}{100}\right]^{0.8} V^2 + \Sigma KV^2 \qquad [15]$$

In these formulas P_c = perimeter of car from plane of top of rails over car to plane of top of rails, ft; L = over-all length of train, ft; V = speed of train, mph; and ΣK = summation of factors ($K_1 + K_2$ etc.) of the various items whose drag depends on other dimensions than perimeter and length. The factors K will be given later.

Similar formulas were given in the article (35) for streamlined trains pulled by separate locomotives and nonstreamlined (standard) trains.

FURTHER SIMPLIFICATIONS

It was shown previously under "Analysis of Test Results" that skin friction is represented by formula [4a] of the V^2 type. The skin-friction drag coefficient C_f, however, is variable, as it depends on the Reynolds number as indicated by formula [5]. If the expression of formula [5] is substituted for C_f in formula [4a] the coefficient for skin-friction resistance becomes constant for any Reynolds number, alike for small models and full-size vehicles, but the disadvantage is that the speed is expressed in the 1.85 power.

On the basis of our tests we know that on theoretical grounds the full-scale resistance should be expressed by formula [9]. Later it was shown that this can, with a great amount of exactitude, be reduced to a V^2 formula of the type of formula [4]. The error which is being made in this case is negligible, as was shown in Figs. 19 and 20 and in Table 1.

However, if we wish to use formula [4a] and disregard the inconvenience of the variation in coefficient C_f, we could easily do so. The variation in C_f is shown in Fig. 19. Reynolds' numbers for models, as already stated, fluctuate between 2,100,000 and 14,250,000, the logarithms of which vary between 6.322219 and 7.153815. For full-size equipment, Reynolds' numbers vary between 88,200,000 and 304,000,000, of which the logarithms are 7.945469 and 8.482874. Thus, it is easily seen from Fig. 19 that, while for models C_f is represented by the upper branch of the curve and fluctuates between 0.0032 and 0.0042, for full-size equipment, the skin-friction coefficients fluctuate only between 0.0020 and 0.0024; in other words, this variation is not very great. The error which would be made by using an average for the latter range is still less, because the second member of formula [9] is small compared with the first; K_f being smaller than K_p, and $V^{1.85}$ numerically smaller than V^2. At 100 mph $V^{1.85}$ is about 50 per cent of V^2. This explains the smallness of error when the two-member formula [9] is converted into formula [4], and this is reflected in Figs. 19 and 20.

Now, if the skin-friction V^2 expression, as given in formula [4a] should be used with the approximation of coefficient C_f, the wetted surface of the car will be in proportion, of course, to the length. Thus, the length should appear in the first power, and whatever theoretical justification for having the length in the form of an exponential function may be, the error may be compensated by varying the skin-friction coefficient, when the train-resistance coefficient is assumed to follow the straight line.

On the basis of these speculations and guided by the desire to follow the law of proportionality referred to previously, the author thought that he should reconsider the exponential curves for trains and cars.

Fig. 21, which shows the two curves for open and closed skirts from the DeBell-Lipetz article (35) proves that the major part of the dashed curves, where the length of the train consisting of

cars alone (without the power car), as tested, comes into play, differs very little from the solid-line curves. Both curves and straight lines are plotted multiplied by P_c = 29.33 ft. Test points for one, two, three, and four cars between the power car and tail are also given. It can be seen that the upper straight line represents the test points just as well for cars with open skirts as the curved line and, therefore, the author suggests that, instead of formula [14] (for open skirts), the following formula for fully streamlined power-car trains with open skirts, be used

$$R_a = (0.00112A_c + 0.00136P_c \frac{L - 150}{100} + \Sigma K) V^2 \quad [16]$$

where A_c = frontal (cross-section) area of the train, sq ft; P_c = perimeter of cars from plane of top of rails over car to plane of top of rails on the other side, ft; L = over-all length of train, including the front (power) car, ft; and V = speed of train, mph. The values of $\Sigma K(K_1, K_2, K_3$, etc.) given in that article (35) remain unchanged. For Fig. 2 all K's were supposed to equal 0.

No drag-test values for trains longer than that shown in Fig. 2 (410 ft) were observed, and the branch of the exponential curve beyond this length, up to 1000 ft, was obtained by extrapolation. In other words, there is no good reason why the straight-line formula [16] should not be taken as justified by test. Furthermore, the difference between these and the exponential formulas for a train length of 1000 ft, does not exceed 6 per cent; i.e., the questionable portions of the curves are not worth considering.

Likewise, for closed skirts the coefficients of the formula should be reduced, and the formula will thus be

$$R_a = (0.001A_c + 0.00121P_c \frac{L - 150}{100} + \Sigma K)V^2 \quad [17]$$

The first members of formulas [16] and [17], depending upon A_c, represent the head resistance, while the second members, with P_c, are mainly skin-friction resistances. At L = 150 ft, these members become zero, and strictly speaking, they are only correct if the length of the power car and tail car is the full-size length of the tested model, which was 142.2 ft. These members should have been thus $0.00136P_c$ (L — 142.2/100) and $0.00121P_c$ (L — 142.2/100), respectively, but the errors by making them as suggested in the above formulas are negligible.

As the power car with tail car cannot be much shorter than 150 ft, the formulas can be applied to any power car, rail car, or train. If they are shorter, the numerators in formulas [16] and [17] become negative. This is correct, because the skin friction of the car is less. However, the formulas should not be applied to cars shorter than 100 ft. For such, only the first member, with A_c, should be used.

Regarding the different corrections K, ΣK is the summation of factors ($K_1 + K_2$ + etc.) of the various items of which the drag depends on other dimensions than perimeter and length. The values of K are:

K_1 = factor for the power-car nose shape shown in Fig. 3:
- For nose bluntly streamlined (No. 2)
  - K_1 = +0.000036 × cross-sectional area of nose at full section, including trucks in sq ft
- For nose well-streamlined (No. 1)
  - K_1 = 0

K_2 = factor for the tail shape shown in Fig. 5:
- For tail bluntly streamlined (No. 2)
  - K_2 = +0.000061 × cross-sectional area of tail at full section, including trucks in sq ft
- For tail well-streamlined (No. 1)
  - K_2 = 0

The length of the tail car should be included in the total length of the train.

K_3 = factor for power-car trucks:
- For two faired trucks
  - K_3 = 0
- For two unfaired trucks
  - K_3 = +0.00026

K_4 = factor for fairing trailing-car trucks on streamlined trains:
- For faired trucks
  - K_4 = —0.00013 × number of trailing-car trucks
- For unfaired trucks
  - K_4 = 0

K_5 = factor for diaphragm shape:
- For smooth diaphragms
  - K_5 = 0
- For corrugated diaphragms
  - K_5 = +0.000037P_c × number of diaphragms

K_6 = factor for bulge of power car:
- For no bulge (as tested)
  - K_6 = 0
- For a bulge of good streamline shape
  - K_6 = +0.00032 × cross-sectional area of bulge in sq ft
- For a bulge of relatively poor streamline shape
  - K_6 = +0.00051 × cross-sectional area of bulge in sq ft

The streamline-train tests were made with a power car, tail car and a number of streamlined coaches. For locomotive trains, tests were made with the locomotive and tender alone, without tail car, and locomotive with tail car, but always without coaches, since streamlined-train data can be used for the resistance of cars in a locomotive train as established by the foregoing formulas.

Regarding streamlined steam locomotives, values of locomotive resistance with different noses, boilers, and shrouds are given in Table 1. For practical purposes it is advisable to consider the resistance of the best streamlined locomotive as a basis and correct this figure, as it was done with the cars of the streamlined train, by adding the differences K due to the omission of one or another feature of good streamlining. The lowest resistance was found for the streamlined locomotive with helmet nose, round-top boiler, and long (closed) shroud (test No. 27), namely

$$R_{al} = 0.1275 \approxeq 0.000924A_L$$

where A_L is the frontal area of the streamlined locomotive, which is 137.98 ≊ 138 sq ft.

Other locomotive corrections are:

K_7 = factor for wheel shrouds on streamline locomotives shown in Fig. 7:
- For closed shrouds (all wheels completely enclosed), K_7 = 0
- For open shrouds shown in Fig. 7 (2 ft × 2 ft 6 in. inspection openings over the driving-wheel journals), K_7 = +0.0005 × total number of openings
- For short shrouds shown in Fig. 7 (driving wheels and tender trucks completely exposed), K_7 = +0.0182

K_8 = factor for nose shape on streamline locomotive shown in Fig. 7 and 7A:
- For helmet nose, K_8 = 0
- For straight nose, K_8 = +0.0021
- For round nose, K_8 = +0.0026

K_9 = factor for boiler shape on streamline locomotive shown in Fig. 7:
- For round top, K_9 = 0
- For cowled top (domes and fittings enclosed in longitudinal cowl above boiler shroud), K_9 = +0.0035

On the basis of the foregoing, the formula for a streamlined train consisting of cars with open skirts and driven by a streamlined locomotive, can be represented by a formula

$$R_a = (0.000924A_L + 0.00136\, P_c \frac{L - 100}{100} + \Sigma\, K)V^2 \quad [18]$$

If the streamlined cars of the train have closed skirts, the formula should be

$$R_a = (0.000924A_L + 0.00121\, P_c \frac{L - 100}{100} + \Sigma\, K)V^2 \quad [19]$$

In formulas [18] and [19], A_L is the frontal area of the locomotive. Other notations are the same as before. For coefficient 0.000924 see Table 1, test No. 27.

Similar to what was said regarding formulas [16] and [17], the second members in brackets of formulas [18] and [19] have the subtrahends in the numerators equal to 100, as a round figure, because the length of the tested locomotive and tender was 96 ft, and the error is slight, as compared with what we would have had if the subtrahends were taken equal to 96. If the locomotive is longer, or shorter, the adjustment will be made automatically when the correct length of the locomotive will be included in the total length L of the train.

Irrespective of the subtrahends of the members in the brackets, 100 or 150, the increase of the air-resistance coefficient per 100 ft of train length is always the same. For streamlined cars with open skirts the increase is, according to formula [18]

$$0.00136P_c \quad [20]$$

and for streamlined cars with closed skirts, it is

$$0.00121P_c \quad [21]$$

The perimeter of streamlined cars P_c usually varies between 28 ft 6 in. and 31 ft 6 in., the average being 30 ft. For our test cars, P_c was 29.33 ft. For open skirts, formula [20] will give at $P_c = 30$, or

$$0.00136 \times 30 = 0.0408$$

and formula [21] will give for closed skirts

$$0.00121 \times 30 = 0.0363$$

Thus, a very simple approximate rule can be established for air resistance R_a of streamlined trains. The rule is

$$R_a = KV^2 \quad [4]$$

where K is the general locomotive and train air-resistance coefficient, equal to the sum of K's for locomotive and cars, and V is the speed, mph. For the locomotive, K should be taken from Table 1 (tests Nos. 21–27), depending upon the nature of streamlining. For every 100 ft of the length of cars 0.0408 should be added for open skirts, while 0.0363 should be added for closed skirts.

For instance, if we have a train consisting of a locomotive 100 ft long, with helmet nose, round-top boiler, and short shroud, uncovering the drivers, and of four streamlined 70-ft cars with open side-sill skirts, the coefficient for the whole train K_t, will be (from test No. 26)

$$K_t = 0.1575 + (0.0408 \times 2.8) = 0.1575 + 0.1142 = 0.2717$$

and the resistance formula is

$$R_a = 0.2717V^2$$

If the locomotive is longer than 100 ft and a more accurate figure is desired, the excess length should be added to length of the four cars. If there is a bulge between the tender and the first car, or if there is a tail car, corrections K should be added or deducted, as the case may be. No corrections for locomotive noses, boiler tops, or shrouds (K_7, K_8, and K_9) should be made, as this is already taken care of in the K of the locomotive in Table 1. If the perimeter of the cars P_c is known, and a more accurate figure is desired, the increase in air-resistance coefficient per each 100 ft of train length can be taken as $0.00136P_c$, or $0.00121P_c$, for open or closed skirts, respectively. Likewise, if the frontal area of the locomotive A_L is known, instead of K, a should be taken from Table 1 and the locomotive coefficient should be figured. Also in this case corrections K_7, K_8, and K_9 should be disregarded.

For standard locomotives and trains it was not so easy to find a consistent law, probably for the reason that the gaps between the cars introduced eddies and turbulence which cannot be represented by one formula. Furthermore, it should be said that models of standard equipment, which were used at our tests at New York University, were of two different scales ($^1/_{16}$ and $^1/_{32}$) and the two-member formulas for these equipments, when expanded to full scale, did not give identical results. This can be seen from Fig. 22, where points representing results of tests Nos. 28 to 35, inclusive, are shown on a chart drawn to a larger scale.

The method of expanding the test results to full scale, which was so helpful for the study of the test results with streamlined equipment, was evidently inapplicable to standard equipment. The power of 1.85 of the second member in formula [9] probably ought to be increased, coming closer to 2, which represents turbulent air flow better than 1.85. In our standard equipment, which has air gaps, turbulent flow probably predominates, and the extrapolation should be made more in relation to the square of the size S_c^2 than $S_c^{1.85}$.

In view of this, the author again analyzed test sheets Nos. 28 to 35, inclusive, which pertain to tests with $^1/_{16}$- and $^1/_{32}$-scale models of the standard locomotive and cars. The fifth column of the test sheets, that is, Tables 2, 3, 4, and 5, gives the net drag, from which the side skin friction (sixth column) was deducted in order to obtain the form pressure, which we know varies directly with V^2. In view of this speculation, the figures of the seventh column (D_p) were now divided over the square of the speed and the square of the scale size, either $V/16$ or $V/32$ and the values of $D_p/V^2S_c^2$ compared. The averages of the values for different speeds are shown in Table 8, together with values K and designation of models from Table 1.

Figures of $D_p/V^2S_c^2$ from Table 8 are also marked on Fig. 22. From what was stated previously, the actual curve of air-resistance coefficient should lie between these values and the values of K. Some of the points do not seem to be very consistent in their relation to each other. Therefore, it is impossible to state definitely whether for air resistance of cars the straight-line law does, or does not, apply. If this inconsistency were discovered earlier, we might have repeated the tests with small models, but as the object of the tests was not so much to determine the formulas for standard equipment, as for streamlined railroad equipment, and as the results, as can be seen from Fig. 22, are very close anyway, we have drawn a compromise exponential curve which satisfies all test data and is in conformity with theoretical speculations referred to previously.

The curve for the standard cars, shown in Fig. 22, corresponds to a formula

$$K_c = 0.00283P_c \left(\frac{L_c}{100}\right)^{0.80} \quad [22]$$

where P_c, the perimeter of our test cars, is 37.83 ft, and L_c is the length of the car portion of the train.

TABLE 8 COMPARISON FOR $D_P/V^2S_c^2$ FOR $^1/_{16}$- AND $^1/_{32}$-SCALE MODELS

Designation of model[a]	For $^1/_{16}$-scale model		For $^1/_{32}$-scale model	
	K	$D_P/(V/16)^2$	K	$D_P/(V/32)^2$
N.L.	0.2611	0.2747	0.2547	0.2725
N.L. + 1 N.C.	0.3420	0.3631	0.3343	0.3612
N.L. + 2 N.C.	0.4124	0.4400	0.3933	0.4360
N.L. + 3 N.C.	0.4486	0.4826	0.4593	0.5026

[a] Model designations are the same as in Table 1.

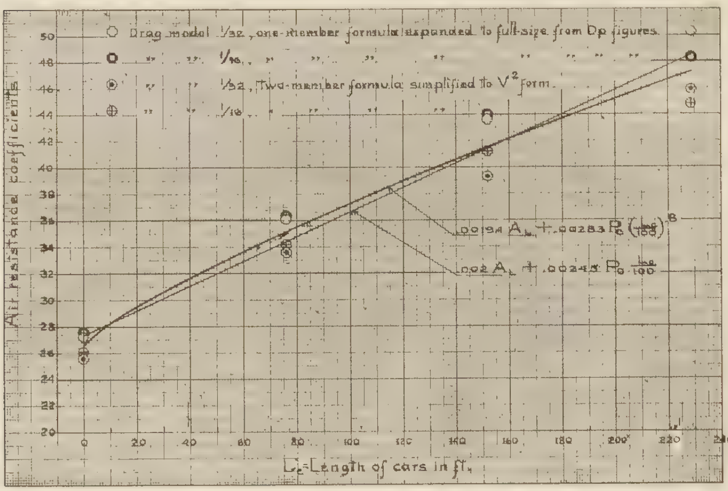

FIG. 22 AIR-RESISTANCE COEFFICIENTS OF STANDARD TRAINS

It is evident from Fig. 22 that a straight line corresponding to a formula

$$K_c = 0.00245P_c\left(\frac{L_c}{100}\right) \quad [23]$$

would still be within the region of the quadruple points and, according to what was stated previously, could be justified as a coefficient for standard car resistance.

Regarding the intersection of both the curve and straight line with the vertical ordinate line corresponding to zero length of cars, the first was taken as an average of K's in Table 8 for locomotive and tender alone, which is $K_L = 0.266$, corresponding to $a = 0.00194$, and the second to $K_L = 0.274$, corresponding to $a = 0.002$, which is very close to the foregoing and is exactly the coefficient given by Cole in the Locomotive Handbook (8), p. 20.

Thus, for standard locomotive and standard cars either of the following two formulas can be applied

$$K = 0.00194A_L + 0.00283P_c\left(\frac{L_c}{100}\right)^{0.80} \quad [24]$$

and

$$K = 0.002A_L + 0.00245P_c\left(\frac{L_c}{100}\right) \quad [25]$$

where A_L = frontal area of the locomotive, sq ft; P_c = perimeter of the car from plane of top of rails over car to plane of rails on the other side of the track, ft; and L_c = length of the car portion of the train, ft.

Therefore, the air resistance of the standard train is

$$R_a = \left[0.00194A_L + 0.00283P_c\left(\frac{L_c}{100}\right)^{0.80}\right]V^2 \quad [26]$$

or

$$R_a = \left[0.002A_L + 0.00245P_c\left(\frac{L_c}{100}\right)\right]V^2 \quad [27]$$

In the opinion of the author, either of the two formulas has the same justification and he would not recommend the extrapolation for longer trains of either of the two until further wind-tunnel tests or data from actual tests are available. Formulas [25] and [27] have the advantages of being simpler, and in the light of the foregoing speculations, these two formulas should be given preference.

Conclusion

Formulas [16], [17], [18], [19], and [27] are recommended by the author of the paper for air resistance of streamlined, power-car, and locomotive-driven trains, and trains of standard cars driven by standard locomotives.

Full-Size Tests

The question arises, of course, how do all these formulas, derived from wind-tunnel tests, agree with actual service results. In order to make such a comparison, it would be necessary to have tests in which the air-resistance effects would be separated from other items of train resistance. Very few tests of this kind have, however, been made.

In the early part of this century high-speed tests were made on a special straight piece of track of the Military Berlin-Zossen Railway, with electric and steam locomotives. The tests were conducted during the summer months of 1903–1905. The maximum speed of these tests was supposed to be 124 mph but actually speeds up to 128 mph were attained. Data of very great value were then secured on the question of air resistance. For instance, it was found that if a car were running at 50 mph the maximum air pressure at the front of the car would be about 7 lb per sq ft, but if the speed were doubled, the pressure would be four times as great. When the car was properly shaped, the resistance figures were reduced (36).

During these tests hydraulic-pressure (pitot) tubes were placed in holes bored at different points in the front wall, as well as in the rounded and beveled sides of the car. According to these measurements, which have been made up to 93 mph, the air resistance satisfied the following formula

$$p = 0.00285AV^2$$

where p = frontal air pressure, lb per sq ft; A = frontal area of the locomotive, sq ft; and V = speed of the train, mph (37).

This formula gives higher results than our formula for standard (nonstreamlined) locomotives, found from test No. 28 (Table 1).

$$R_a = 0.001906AV^2$$

Before the Louisiana Purchase Exposition, St. Louis, 1904, a special electric testing car, "Louisiana," was constructed for the purpose of determining the effect upon the front, the sides and the rear of a car when running on tangent level track at high speeds. Vestibules were designed and constructed in such a way as to permit determining the effect of the shape of the vestibules and also to separate the vestibule pressure from the total car-body pressure. This was achieved by making the front and back walls of the vestibules movable, so that the head-on pressure and suction pressure could be directly measured by dynamometers of the knife-edge scale type built by Fairbanks, Morse & Company. As the electrical input data were known, the separation of the total power into its various components was possible. The car, constructed by the J. G. Brill Company, had a car body 32 ft long, exclusive of vestibules, and was placed on a 100-ft flat car so that the vestibules could move on rollers.

Different shapes of vestibules were provided for testing: (*a*) standard, rounded; (*b*) a special, flat; (*c*) a special parabolic, and (*d*) a special parabolic-wedge shape. The rear vestibules were also made of four different shapes, and they could be interchanged.

A detailed description and results of the tests were given in the report presented by the Electric Railway Test Commission to the president of the Louisiana Purchase Exposition (38). Observation points were obtained directly up to 55 and 60 mph and later extrapolated to 100 mph. The parabolic wedge showed less head-on pressure than the parabolic shape; for suction it was reversed. The greatest head-on pressures and suctions were received, of course, from the standard and flat surfaces. For practical purposes it was found more advantageous to use the parabolic wedge for the front and the parabolic shape for the rear. As compared with the standard and flat surfaces, the resistances in lb per sq ft were found to be as follows:

	50 mph	100 mph
Parabolic wedge in front and parabola in the rear	1.52	7.25
Flat in front and in the rear	5.97	23.45
Standard in front and in the rear	3.88	17.08

Thus, the resistances did not follow exactly the V^2 law, but taking into consideration the difficulties in making those tests, the influence of friction of the moving parts and other predicaments, it would seem that the general increase of resistance was sufficiently well in proportion to the square of the speed. The coefficients, although different from ours, are close and of the same order.

Figures for the power absorbed by the front and rear vestibules are also given in the report (38). They are in proportion to the cube of the speed, as should have been expected.

The Berlin-Zossen tests brought about the desire of accelerating trains. Steam locomotives on some railroads had been equipped with wind cutters, usually cones on the smokebox. These were used on the P.L.M. in France, on the Bavarian Railways in Germany, in Hungary, and on some other European railroads. They showed a saving of 100 hp[5] in wind resistance at a speed of 100 klm per hour (about 62 mph). Nevertheless, streamlining did not make much progress in Europe at that time; neither were greater results achieved in this country, except only that some changes had been made in the shaping of the vestibules on some electric cars. It is interesting to note that in the concluding chapter, p. 574, of the Electric Railway Test Commission's Report (38) the following statement was made:

> "The results of these tests are not only applicable in connection with electric cars but they also yield many suggestions regarding the construction of steam locomotives and trains. .... The resistance offered to all high-speed steam trains by the air would be materially diminished if the locomotive were housed in a shell and if pains were taken to remove all unnecessary projections from the coaches and, further, if the outline of the tender were made to conform to the general cross-sectional shape of the train."

The suggestion of putting the locomotive in a shell, now called a streamlining shroud, had to wait 30 years until it was first materialized in the Hiawatha and other steam locomotives, both in this country and in Europe.[6]

Very interesting tests were recently made by Parmantier on the French P.L.M. Railway. In 1935 this railway streamlined in their shops a 30-year-old 4-4-2 locomotive which they rejuvenated by installing a superheater, a feedwater heater, and a high-efficiency P.L.M. exhaust nozzle. This engine has since been pulling high-speed, lightweight trains in regular service, but before putting it in operation, Parmantier made very exhaustive tests, with the object of studying streamlining. Before testing, this locomotive was calibrated at the new stationary testing plant at Vitry, France, together with a nonstreamlined locomotive, by determining the power which the two engines were able to deliver at 60, 80, 100, 120, and 140 klm per hr with varying throttle and cutoffs. The two locomotives were identical, but one of them was streamlined and the other was not. For the eventuality that the locomotives had different axle frictions, the calibration of power of the engine part was made separately for the streamlined and nonstreamlined locomotives.

Afterward the two locomotives were put in service, pulling trains of four standard and of four streamlined cars, trying to duplicate the performance at the testing plant, and observing the variables (speed, cutoff, and opening of the throttle). Thus, the power of the standard and streamlined locomotive with standard and streamlined cars was ascertained and the gain in power due to streamlining noted. The last car of the streamlined train had a partly streamlined tail.

For the streamlined locomotive with the streamlined four-car train, the saving in power was found to be represented by a formula (40)

$$P_m = 0.000163V^3$$

where P_m represents the saving in metric horsepower, and V is the speed, klm per hr. In English units this corresponds to

$$P_s = 0.00067V^3 \quad \text{[28]}$$

where P_s is the saving in British horsepower and V is the speed, mph.

On the basis of our full-scale simplified formulas, the resistance of the French trains should be calculated as follows: A_L = 126.3 sq ft; P_c = 32.5 ft; $L_t = L_l + L_c = 70 + 300 = 370$ ft; and L_c = 300 ft.

Formula [18] will give, for the streamlined French train

$$R_a = (0.000924 \times 126.3 + 0.00136 \times 32.5 \times 2.7 + \Sigma K)V^2$$
$$= (0.1167 + 0.1193 + \Sigma K)V^2 = (0.2360 + \Sigma K)V^2 \text{ lb}$$

Regarding corrections K for the French streamlined train, only K_2, K_4, and K_8 should be considered, namely:

$$\begin{aligned} K_2 &= 0.000061 \times 126.3 &&= +0.0021 \\ K_4 &= 0.00013 \times 8 &&= -0.0010 \\ K_8 &= &&= +0.0021 \\ & \qquad\qquad \text{Total} &&= +0.0088 \end{aligned}$$

Thus, K = 0.0088, and the foregoing formula becomes

$$R_a = (0.2360 + 0.0088)V^2 = 0.2448V^2 \text{ lb}$$

For the standard French train, formula [27] will give

$$R_a = (0.002 \times 126.3 + 0.00245 \times 32.5 \times 3.0)V^2 = 0.4914V^2$$

The saving in air resistance due to streamlining of the whole train is

$$0.4914V^2 - 0.2448V^2 = 0.2466V^2 \text{ lb}$$

The gain in power will thus be

$$\frac{0.2466 \times V^2 \times V}{375} = 0.00066V^3 \quad \text{[29]}$$

This is in very good agreement with formula [28]; in fact, a

[5] A rather doubtful figure although very often quoted, especially in France.

[6] In this connection it is interesting to note that a U. S. Patent, No. 490,057, was issued to F. U. Adams on January 17, 1893, on Locomotive and Tender Housing. F. U. Adams also published a book in 1892 on Atmospheric Resistance (39).

smaller figure is rather gratifying, for reasons which will be given later. Parmantier states that the saving actually amounted to 450 metric hp at a speed of V = 140 klm per hr, which is equivalent to 443.8 British hp. Formula [29] gives at 87 mph (140 klm per hr) 434.6 hp, which is 3.4 per cent less.

Parmantier also states that the saving in power due to streamlining of the locomotive, which was attached to a nonstreamlined train, was 270 hp, while the streamlining of the cars pulled by the nonstreamlined locomotive amounted to only 90 hp (41). These figures are somewhat inconsistent with the total saving given by Parmantier and, therefore, the author is gratified that his (the author's) formulas give smaller saving for the whole train than the P.L.M. figure of 450 hp, which may be slightly exaggerated, especially when we consider that other experimenters also obtained lower figures. For instance, the Northern Railway of France found that at 140 klm per hr on the new Super-Pacific streamlined locomotive, 170 hp was gained by streamlining the locomotive alone, which is less than the P.L.M. figure of 260 hp for the locomotive at the same speed. At 150 klm per hr (93.2 mph) the gain was 200 hp (42). Nordmann also reported figures of actual savings on German streamlined locomotives of the 03 class, and they were also in the neighborhood of 200 hp at 93 mph (43).

Very recently the Paris-Orleans-Midi Railway streamlined one of the Pacific modified Chapelon locomotives and made tests to determine the saving in power. The measuring of the saving was made directly by pushing at one time a streamlined and, at another time, a nonstreamlined locomotive, by an ordinary locomotive and a dynamometer car in between. The difference in the pushing efforts in both cases represented the gain due to streamlining.

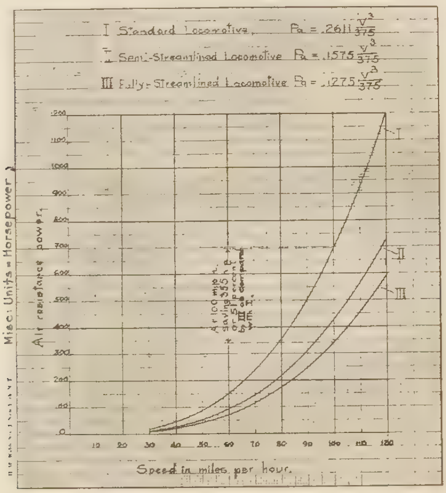

FIG. 23 AIR-RESISTANCE POWER CURVES FOR LOCOMOTIVES

The tests were made in the summer of 1937, and the following figures were obtained:

Speed in klm per hr	Actual gain in British hp	Gain in British hp according to formulas of this paper
90	59.2	57.1
100	79.0	78.4
110	103.1	104.3
120	128.0	135.5
130	147.6	172.2

Thus, the agreement at all speeds, except the last, is very good. Furthermore, the difference may be due to the fact that the locomotive was not fully streamlined. If the gains at all other speeds up to 120 klm per hr are correct, the last must be also in the neighborhood of 170 in order to satisfy the parabolic law.

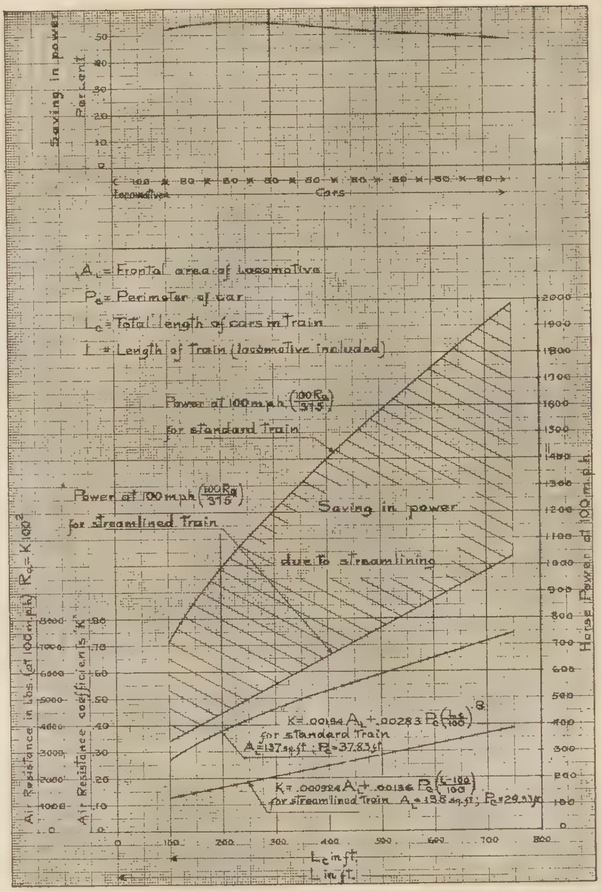

FIG. 24 AIR-RESISTANCE, POWER AND ECONOMY CURVES FOR STREAMLINED AND STANDARD TRAINS

All these data tend to prove that the formulas of the paper at least for locomotives, cannot be far from the truth.

SAVING IN POWER

The advantage of streamlining lies in the saving of power due to decreased air resistance. On the basis of the foregoing results it is easy to figure the amount of saving due to every single improvement in shaping the locomotive and cars. Figs. 23 and 24 show curves representing the power which can be thus saved by streamlining locomotives alone, and locomotives with cars. The curves are self-explanatory. For Fig. 24, formula [26] was used for the standard train, since this gives a slightly smaller and more probable figure for saving.

A passenger car or a Diesel-power train, which are totally enclosed, can be shaped so as to result in the greatest possible power saving. A steam locomotive, unfortunately, has to be partly exposed for the accessibility of parts for inspection. In the construction of streamlined locomotives the two conflicting demands of greatest possible saving and accessibility are always militating against each other and the actual design is, therefore, a compromise between the two.

Fig. 25 shows the "Hiawatha" locomotive for the Chicago, Milwaukee, St. Paul & Pacific, the first streamlined locomotive built by the American Locomotive Company, and Fig. 26 shows the Gulf, Mobile & Northern streamlined train, the first built by

FIG. 25 *Hiawatha* 4-4-2 STREAMLINED LOCOMOTIVE OF THE CHICAGO, MILWAUKEE, ST. PAUL & PACIFIC RAILWAY

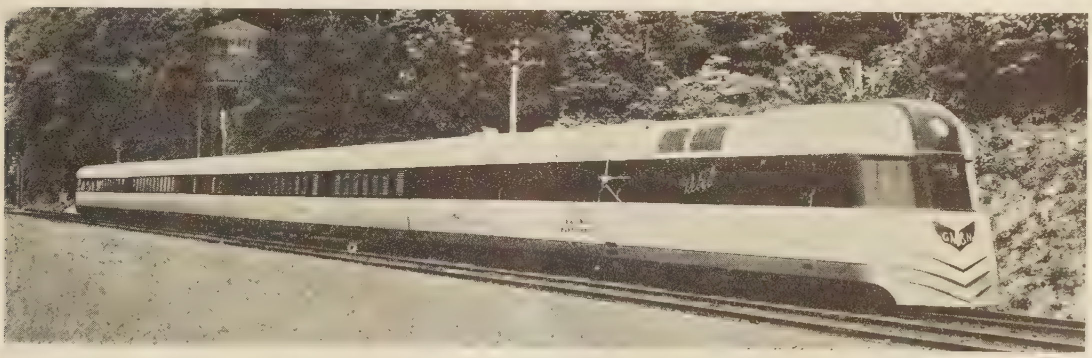

FIG. 26 GULF, MOBILE & NORTHERN STREAMLINED TRAIN *The Rebel*

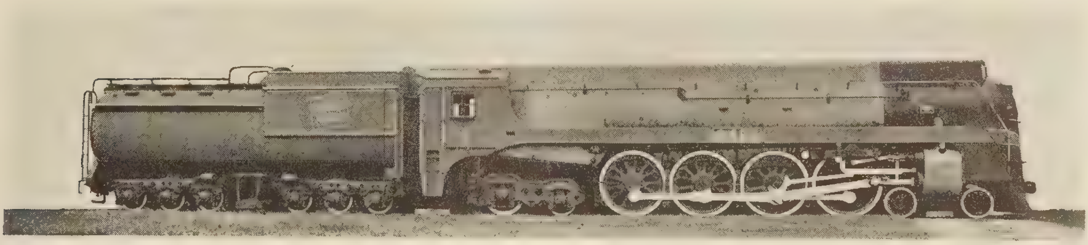

FIG. 27 CANADIAN NATIONAL SEMISTREAMLINED 4-8-4 LOCOMOTIVE

FIG. 28 CANADIAN PACIFIC SEMISTREAMLINED 4-4-4 LOCOMOTIVE

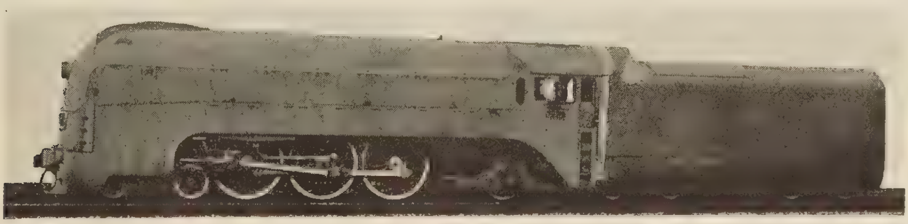

FIG. 31 STREAMLINED 4-6-2 LOCOMOTIVE OF THE SOUTH MANCHURIAN RAILWAY

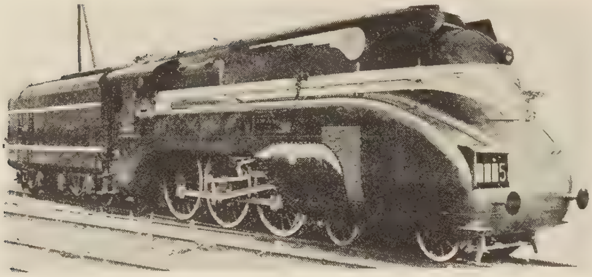

Fig. 29 Streamlined 4–6–2 Locomotive of the Northern Railway of France

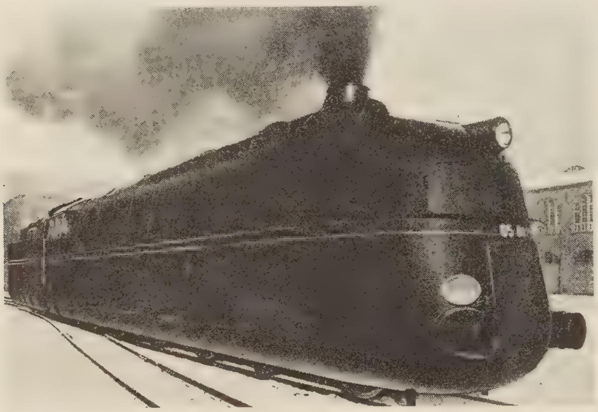

Fig. 30 Fully Streamlined 4–6–4 Locomotive of the German Reichsbahn

(Fig 31 See page 21)

Fig. 33 Pennsylvania Streamlined 4–6–2 Locomotive

Fig. 32 Southern Pacific 4–8–4 Semistreamlined Locomotive

Fig 34 New Haven 4–6–4 Semistreamlined Locomotive

Fig. 35 London, Midland & Scottish Streamlined 4–6–2 Locomotive *Coronation*

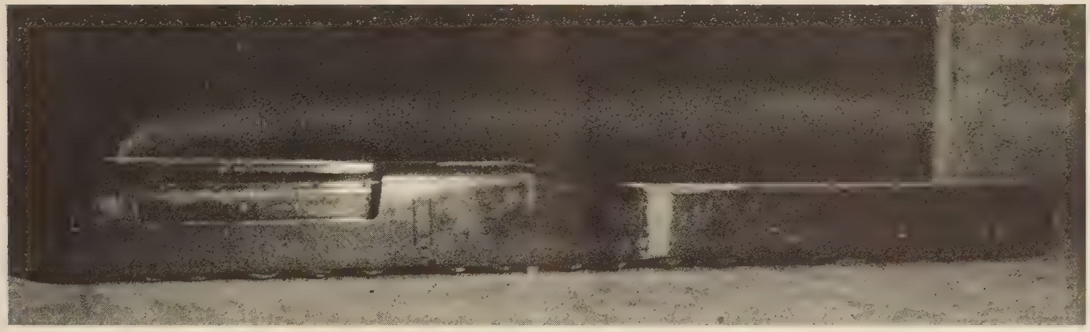

FIG. 36 WIND-TUNNEL SMOKE TEST—MAXIMUM RELATION OF AIR VELOCITY TO EXHAUST STEAM PRESSURE

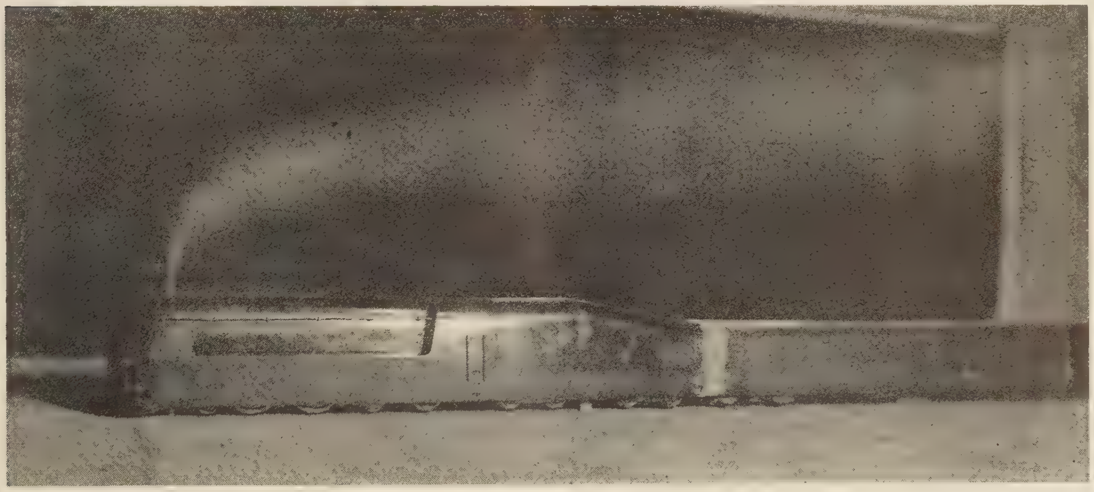

FIG. 37 WIND-TUNNEL SMOKE TEST—MINIMUM RELATION OF AIR VELOCITY TO EXHAUST STEAM PRESSURE

the American Car & Foundry Company—both on the basis of the tests made at the New York University wind tunnel (30). Figs. 27 to 35, inclusive, show the different ways in which streamlining is being accomplished in locomotives of different countries.

SMOKE TESTS

The question of shaping the front part of the boiler and smokebox so as to avoid any interference with carrying away the smoke is also a very important problem in streamlining. Some thought was given to this question at the time of the wind-tunnel tests conducted by the American Locomotive Company. Special smoke tests were made to determine the speeds and exhaust pressures at which the resulting force of the exhaust is sufficient for carrying away the smoke. Figs. 36 and 37 are views of models taken at different ratios of steam and air velocities. The tested arrangement has been incorporated in the "Hiawatha" locomotive and has proved to be very successful in service.

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Discussion

Comparative Torque and Horsepower Requirements of Standard Four-Flute and Spiral-Flute Taps[1]

O. W. Boston[2] and W. W. Gilbert.[3] The results of tests similar to those undertaken by the author have long been needed, even though some tests have been made by the Germans who have been interested in this field of engineering. The writers hope that this work may be continued to cover other sizes of taps, particularly the smaller sizes, which result so frequently in broken tools.

There are a few questions which seem pertinent to a complete understanding of the paper as presented. In the first place from Fig. 2 of the paper the writers believe that a good average value of torque would be of greater value than the maximum torque and average horsepower as was used by the author. Undoubtedly, the maximum torque, as shown in the upper part of Fig. 2 of the paper, is due to the clogging of chips. A good average value of torque would be about 80 per cent of the maximum indicated. The lower part of Fig. 2 of the paper shows the maximum torque when cutting cast iron, which appears to be a good value. However, the writers believe that the horsepower computations based on the area under this curve to give a mean height of the curve, do not represent values pertinent to the cutting operation and that normal maximum values, just as with normal maximum torque, should have been used.

Just below Fig. 4 of the paper the author states that the horsepower was computed from the energy in foot-pounds as determined from the area of these torque-space cards, when in reality this energy in foot-pounds is the average torque for the depth of the hole plus the taper of the tap amounting to eight times the pitch. The writers believe that this value is not definitely tied up with tap performance. This is particularly confusing if one wanted to gain an impression of the power required to tap with plug or bottoming taps, for which there appears to be no means of conversion.

The influence of the sulphurized oil and the lard oil on the maximum torque for various depths of threads and speeds, as shown in Fig. 3 of the paper, is interesting. These values are replotted in Fig. 6 of the paper to show for each depth of thread the maximum torque at any speed. The writers would like to inquire if the author has any explanation as to why, with lard oil, the torque is increased directly with the speed, while the reverse is true when the sulphurized oil is used. Also, the torque when tapping with sulphurized oil is higher in most cases than when tapping under the same conditions with lard oil. The writers have found that this is not true when drilling or turning.

It appears that the legends of Figs. 10 and 11 of the paper are transposed, and that the standard straight-flute tap in the legend of Fig. 10 should be spiral tap, whereas the spiral tap in Fig. 11 should be standard straight. With these changes made, it is seen that the data of Fig. 9 of the paper agree with the same data transposed in Fig. 11 of the paper.

The abscissa of the right-hand portion of Fig. 12 of the paper should, it is believed, be from 0 to 500, rather than from 0 to 100, as it indicates revolutions per minute rather than depth of thread.

Referring to Figs. 12 and 13 of the paper, when comparing horsepower, it is seen that the horsepower for the standard flute tap is greater correspondingly than that for the tap with a spiral-tip flute. Fig. 12 of the paper shows the value of 0.225 hp for the 90 per cent thread at 300 rpm, whereas Fig. 13 of the paper gives a corresponding value of 0.180 hp. On the other hand, Figs. 7 and 8 of the paper showing the maximum torque, give the reverse conclusions in that from Fig. 7 the maximum torque for a 90 per cent thread at 300 rpm is 0.98 in-lb, whereas in Fig. 8 it is 109 in-lb. Possibly this condition is due to the use of maximum torque for recording torque, but an average torque over the length of hole plus taper of tap for computing horsepower.

Max Kurrein.[4] Although this paper gives an excellent comparison of standard four-flute and spiral-flute taps, the writer would like to call to the author's attention several points in the paper which differ from the results of tapping tests published by the writer in 1925.[5] These tests[5] were, as far as the writer knows the first which investigated the cutting forces of taps in actual cutting work. In continuing this discussion it is mentioned that some of the figures in this discussion by the writer relate to a second series of tapping tests made for N.D.I., but the results of which were not published.

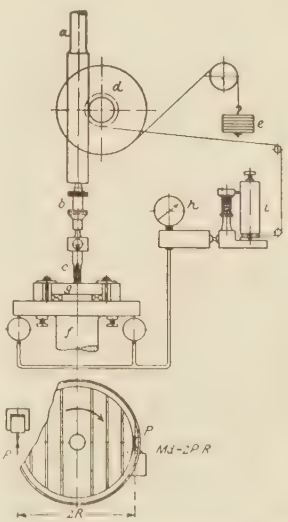

Fig. 1 Torque-Measuring Device With Two Measuring Cylinders Opposite Each Other on the Table

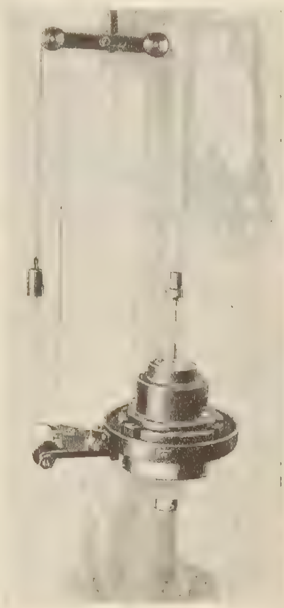

Fig. 2 Device With Axial Spring Bar for Measuring Torque

[1] Published as paper MSP-58-11, by Harry L. Daasch, in the November, 1936, issue of the A.S.M.E. Transactions.

[2] Professor, College of Engineering, University of Michigan, Ann Arbor, Mich. Mem. A.S.M.E.

[3] Instructor, College of Engineering, University of Michigan, Ann Arbor, Mich. Jun. A.S.M.E.

[4] Professor of Mechanical Engineering, Head of Workshops, Technion, Haifa, Palestine. Mem. A.S.M.E.

[5] "Die Prüfung der Gewindebohrer," by M. Kurrein, *Werkstattstechnik*, vol. 19, September, 1925, pp. 601–619.

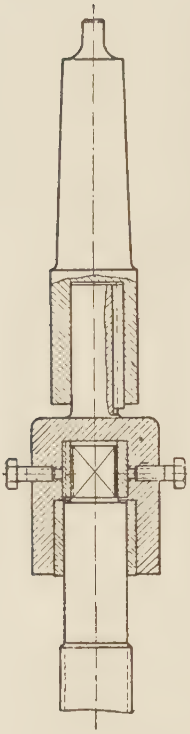

Fig. 3 Special Chucking Device for Holding Taps

The writer does not believe that it is correct to measure the torque by the author's method shown in Fig. 1 of the paper; the writer uses two measuring cylinders opposite each other as shown in Fig. 1 of this discussion in order to measure the pure momentum without an otherwise remaining cross force through the axis of the table. The axial spring bar shown in Fig. 2 of this discussion, which was also used by the writer, is still more limited for measuring pure momentum. The writer considers this very important since in tapping there is no axial thrust as exists when drilling, and consequently the cross force tends to throw the table out of alignment due to the play in the ball bearings. When the exceedingly small depth of tap chips is taken into account, which is 0.0025 in. for a $^1/_2$-in. tap with four flutes and seven-thread chamfer, the necessity of an absolutely true-running tap and table for measuring tests will be appreciated.

For these reasons the writer discarded the usual tapping devices and built the special chucking device shown in Fig. 3 of this discussion. In this device the tap is held with its ground shank in a ground cylindrical bushing, thus insuring absolute alignment; the tap is driven by a loose driver from its square end. With this tap-holding device the writer has succeeded, with taps up to $2^3/_4$ in. in diameter, in cutting threads true without double-cutting.

Fig. 5 Weber Tap With Spiral-Tip Flutes

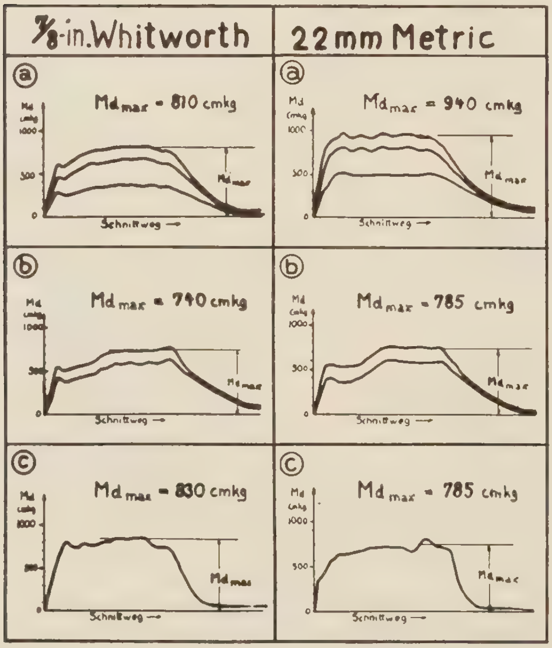

Fig. 4 Torque Developed Using $^7/_8$-In. Whitworth and 22-Mm Metric Taps

[(a) Curves obtained with a set of three taps. (b) Curves obtained with a set of two taps. (c) Curve obtained using only one tap.]

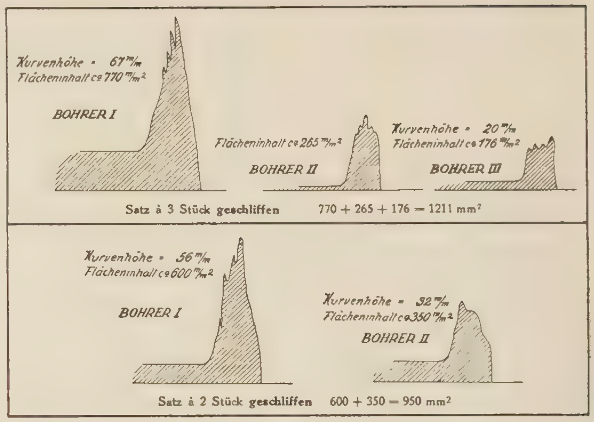

Fig. 6 Test Diagrams of Weber Taps Obtained With the Apparatus Shown in Fig. 2

(Top—Diagrams for a set of three taps. Bottom—Diagrams for a set of two taps.)

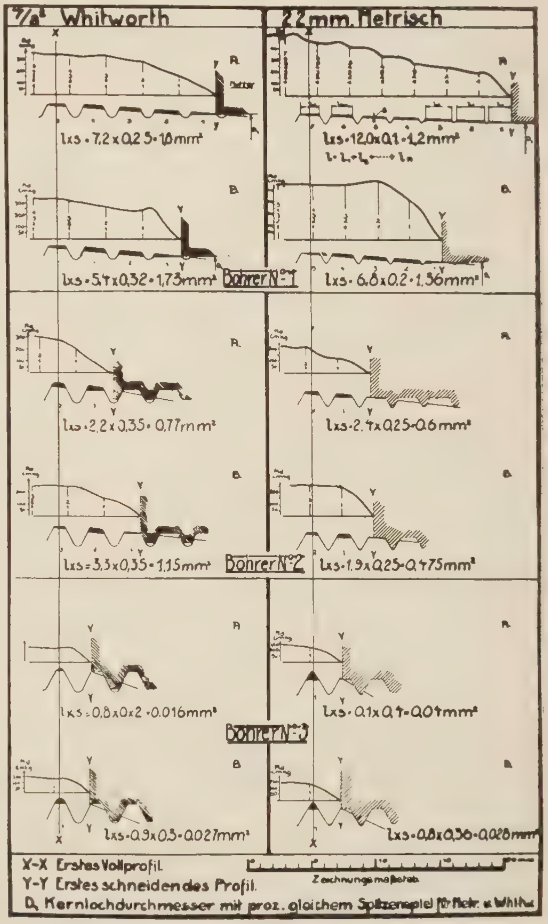

Fig. 7 Cutting Sections of Taps and Forces for Each Cut

The writer observes from Fig. 2 of the paper that apparently the author used standard nut taps (Machinery Handbook, page 1211) with between six and eight threads in the chamfer. When

the writer made his tests, such taps were made with 22 threads in the chamfer. In making the tests for N.D.I. previously referred to, the writer proved that losses due to the long chamfer could be eliminated by using ground taps. Figs. 4*a*, 4*b*, and 4*c* of this discussion show the results of tapping tests when using three taps of a set of three, when using the second and third taps, and when using the third tap only. It will be observed from Fig. 4 of this discussion that the maximum torque Md_{max} remains approximately the same, except with the set of three 22-mm metric taps.

The writer appreciates seeing that the results of his tests have been proved correct by the author.

The advantage of using taps with spiral-tip flutes was proved in 1927 by tests of Weber taps, one of which is shown in Fig. 5 of this discussion. Test diagrams obtained when using but two of these taps, shown in Fig. 6 of this discussion and made on the apparatus shown in Fig. 2 of this discussion, show a gain of $[(1211 - 950)/1211]\ 100 = 21$ per cent approximately, the gain being the result of the new chamfer. The maximum torque shows a gain of $[(115.5 - 88)/115.5]\ 100 = 24$ per cent, approximately.

This can readily be explained by the theory of cutting action wherein the whole cutting action of the tap and all the work consumed in cutting is done by the chamfered portion of the tap, the full threads only guiding the tool. From Fig. 7 of this discussion, showing the actual cutting section, the single chip sections, and the forces for each cut, it is seen that the broad edge of the chamfered profile, representing the main cutting edge, and the short side edges to the right and left of the main edge work together as a formed tool. However, since the main direction of the cut lies at an angle B to the horizontal ($\tan B = h/2\pi d$, where h is the pitch of the thread), one of the side edges in a straight-flute tap does not cut but only scrapes because its breast angle is greater than 90 deg. If then the flute in chamfered portion of the tap is inclined to the axis of the tap, as shown in Fig. 5 of this discussion, thus forming a spiral flute, both side edges will retain their proper breast angle and they will act as cutting edges, thereby decreasing the torque. For the same reason the formed tool for cutting square threads on a lathe is set at right angles to the average pitch angle of the thread.

The writer does not understand why the author developed his horsepower diagrams. The characteristic value for the cut which governs the size of machine to be used is the maximum torque. As the torque varies with the advance of the tap, the integral of the work $\int Pds$ is characteristic for the design of the tap. If the horsepower is substituted, the abscissas are introduced into the ordinates and the characteristic relation (work/speed) becomes obscure.

The writer is surprised at the remarkable increase of the torque with increasing speed as shown by the author. Up until now it has been accepted by many engineers as a result of numerous turning, drilling, and other cutting tests that the cutting force varies but little with the variation of cutting speeds. Still more astonishing is the reversing of this variation when using another lubricant. The writer believes that the viscosity of the lubricants and the increasing temperature at the cutting edge of the tool is responsible for this variation. It would have been interesting if the author had made comparative tests of the taps without using cutting fluid, that is, cutting dry; then the cutting qualities of the two tap designs might have been clearer. This could have been done by registering the full tapping diagrams and observing the friction moment after the first full thread left the underside of the test piece. This is the procedure used by the writer in making tapping diagrams of his tests. The ending of the diagram thus obtained shows the friction between the tap and the thread under actual working conditions, which conditions are very complicated in tapping.

Author's Closure

The suggestions of Messrs. Boston and Gilbert relative to the maximum torque are reasonable but they obviously result in variable answers depending upon the opinion of what might be considered "normal maximum." The author prefers to accept the maximum as shown in Fig. 2 of the paper. The maximum so defined is the value which eventually determines tap breakage. An average torque is obviously secured in the determination of the horsepower and the author believes that efforts leading to other values than this average and the previously mentioned maximum torques will result in confusion and differences of opinions without any compensating gains in further understanding the process.

The author is firm in the belief that horsepower (the rate of doing work) should be reported as in the original paper. Again, it is possible to consider the instantaneous rates of doing work at various periods of the process. Machine-inertia factors make reasonable the use of the average torque in the determination of the horsepower. The writer would caution the reader that the period of increasing and decreasing torques at the beginning and end of the tapping process are dependent upon tap taper and the percentage of thread being cut. Unlimited combination of these items is possible. The author's choice of eight times the pitch is an effort to secure a standard. The establishment of such arbitrary standards is always open to question.

The author is interested in Professor Kurrein's description of this equipment for measuring tap torque. Professor Kurrein may be assured that the author's dynamometer was entirely adequate, sensitive, and accurate for the study reported. Inherent errors due to the equipment were found to be less than 1 per cent of the variations secured in duplicate tappings of the same metals with identical tap and tapping arrangements.

The author was quite as surprised as were the discussers to observe the radical difference in torque trends caused by changes in tapping speeds and cutting fluids. Those are items worthy of further study. The writer appreciates the explanations advanced and he is hopeful that further tests will throw additional light on these phases of the problem.

Development of a Fuel-Injection Spark-Ignition Oil Engine[1]

J. M. Kaderly.[2] The vacuum control for throttle governing the A-C oil engine described by the author consists of a spring-loaded piston actuating a linkage and a control rod which in turn causes the travel of the oil-pump plunger to be increased or decreased as the speed of the engine demands. It appears as though this desired controlling could be done as well, or even better, and certainly much cheaper by means of a cup-type Sylphon bellows assembly.

Considering the diagram shown in Fig. 12 of the paper to be to scale, the control piston is about 1 in. in diameter. If this piston were replaced with a Sylphon bellows of $1^1/_2$ in. outside diameter, a larger area would be obtained than that in a cross section of the piston. The author does not give the necessary travel of the control piston, but if he could arrange his linkage so that a $1^1/_2$ in. Sylphon could be used with a $^3/_8$-in. or even a $^1/_2$-in. total travel, the writer believes the author could replace his present control with an easily attachable cup-type Sylphon assembly about 2 in. in diameter by about 4 in. long. The length of this assembly would depend upon the travel needed. This

[1] Published as paper OGP-59-1, by Nicholas Fodor, in the January, 1937, issue of the A.S.M.E. Transactions.

[2] Assistant Development Engineer, Fulton Sylphon Company, Knoxville, Tenn. Jun. A.S.M.E.

Sylphon bellows could be housed in a heavy brass cup with only one tube connector on the bottom, whereas the author's present design calls for two tube connectors. The Sylphon assembly would not be affected by grease or dirt that would accumulate on any exposed part of a gas or oil engine.

Perhaps the author has already considered a bellows control and found it unsatisfactory. If this is the case the writer would appreciate an explanation as to why it did not prove satisfactory.

O. L. Riegels.[3] The author describes a compromise engine of a type well-known and developed in the Scandinavian countries, one of which is the Penta-Hesselmann engine.

The author has referred to Figs. 1 and 2 of the paper as a comparison of the diagrams of the A-C oil engine and a Diesel engine of similar dimensions. The term Diesel is not justified in the explanation of these figures because the diagrams are those of Otto-cycle engines of different compression ratios. Compared with a Diesel diagram of the same mean effective pressure, the difference in a crank-effort diagram would doubtless be in favor of the Diesel engine. When the author terms one diagram shown in Fig. 1 as a Diesel-engine diagram, it is excusable since it can be defended on the grounds that practically any of the present high-compression oil engines described in literature as direct-injection Diesel engines have this particular diagram, and are conventionally termed "Diesel-cycle engines."

The type of engine adopted by Allis-Chalmers for their tractor is a convenient compromise until the united efforts of the engineers in this field develop a true direct-injection Diesel engine.

The spark-ignited oil engine diverges from the general trend of simplicity in that it embodies a gasoline-engine ignition system as well as a complete fuel-oil injection system, which finally results in increased maintenance and service costs.

The facile conversion of a commercial gasoline engine into a spark-ignited oil engine is naturally tempting, but the low compression ratio and slow flame propagation inherently puts this type of engine in the lower over-all efficiency brackets.

Although the fuel consumption and general results obtained by the A-C engine is very fair to engines of this type, the writer feels that a precombustion engine designed for a low pressure rise (small precombustion chamber and narrow throat opening) would be simpler and more efficient.

Experience indicates, that an oil engine capable of replacing the gasoline engine in all kinds of automotive power plants will not be obtained before a true direct-injection Diesel engine is developed.

A great amount of intelligent research work has been done in the last ten years, and from time to time discloses the behavior of combustion under different conditions; the author's diagram analysis is very valuable in this respect.

The author's method for controlling the quantity of air in accordance with the momentary load is interesting and, as far as the writer knows, its applications to oil engines is new.

The Deco pump described in this paper is a novel and simple design, but as is the case with all pumps of this type for individual cylinders, its pressure varies as the square of the plunger velocity, and therefore is not entirely practical for automotive application.

As a result of more than 30 years experience on fuel-oil injection the writer believes that future research should be concentrated on the development of a gradual injection, whereby the combustion would be controlled and the injection lag decreased to a minimum. Also, for automotive application, a pump system should be developed for delivering the fuel at a constant pressure at all times, preferably about 4000 lb per sq in. except when stronger penetration is desired which pressure seems to be the border line for gain in dispersion. Most probably the best solution will be the redesigning of the common rail system, so this trustworthy system could be adapted to high-speed engines.

[3] Mechanical Engineer, Diesel Engine Development, The Yoder Company, Cleveland, Ohio. Mem. A.S.M.E.

Author's Closure

Referring to J. M. Kaderley's statement that Sylphon bellows could have been used for the vacuum control described in the paper, the author advises that this matter was discussed with the Fulton-Sylphon Company before the use of the piston arrangement was decided upon. However, their requirements for the satisfactory operation of the bellows was such that it made its adaptation impossible.

If, in O. L. Riegels' remarks, it is meant that the Allis-Chalmers engine and the Hesselman engine are of the same type, he is correct, provided his comparison refers to the fuel-injection and spark-ignition features of the two engines. However, as pointed out in the paper, the creation of the mixture and the combustion process proper are entirely different in the two engines.

Mr. Riegels discusses Figs. 1 and 2 of the paper, which show indicator diagrams of the Allis-Chalmers oil engine and a Diesel engine of similar dimensions, and he criticizes them mainly on the basis that the Diesel-engine diagram resembles the Otto cycle too much. However, Mr. Riegels admitted later in his discussion that practically all of the present compression-ignition oil engines have an indicator diagram as shown in these two figures.

The Allis-Chalmers Manufacturing Company did not build the engine described in the paper as a compromise unit to take the place of the Diesel engine because it was believed that the fuel-injection spark-ignition engine has a field of its own. The question has arisen as to whether or not the addition of an ignition system to the injection system increased the complexity of the Allis-Chalmers engine. The author believes this question has been answered by the thousands of users of this type of engine who are of the opinion generally that the maintenance, the operation, and the length of service obtained from it are at least equal to the competitive Diesel engines.

Mr. Riegels is entirely mistaken when he believes that slow flame propagation is inherent with low compression ratios. In this respect there is no difference between the fuel-injection spark-ignition engine and the gasoline engine. The lower fuel economy of the spark-ignition engine is due entirely to the lower compression and expansion ratios as compared with those of a Diesel engine.

The author agrees with Mr. Riegels that there is no oil engine at present for automotive purposes having a characteristic of a true direct-injection Diesel engine, and that it will take years for the development of a gradual injection whereby the combustion will be controlled in accordance with the application of the engine. Whether the pump system for such an engine will be of the constant-pressure type, the author is unable to say at this time.

Mr. Riegels' statement that, in the Deco pump the pressure varies as the square of the velocity of the pump plunger, is not correct, because he did not take into consideration the pressure waves and the elasticity of the fuel, both of which change the pressure rise in the injection system considerably.

Design and Operating Problems With Gas- and Oil-Fired Boilers for Stand-By Steam-Electric Stations[1]

Emil Creutz.[2] Mr. Estcourt's paper on the design and operating problems for stand-by steam-electric stations is undoubtedly a most interesting and valuable contribution to the literature on this type of plant and the data and information regarding the boiler problems of such plants are most helpful.

In his discussions of the problems, however, Mr. Estcourt, in his paper, devotes most of his attention to the problems of boiler operation. Also, the plants discussed are confined to those of the type in which the turbines are kept in operation between the peak-load periods or other conditions which call for capacity operation.

In connection with this problem of operation of stand-by plants, it may be of interest to consider the problems involved in turbine operation when full load in the shortest possible time is required, particularly in the case of plants where the general factors governing the operation of the plant make it impractical or undesirable to maintain the turbine in operation between the peak-load periods or times when the stand-by capacity of the plant is required.

In connection with this phase of stand-by plant operation, the double-rotation radial-flow turbine is peculiarly well-suited for its ability quickly to pick up load. The masses of this type of turbine for a given size are comparatively small and the symmetrical nature of the blade system, in so far as expansion factors are concerned, permits the turbine to be subjected to full steam pressure and temperature even when the turbine is cold. This ability in practice means practically complete elimination of the warm-up period as this period is usually thought of, and applies not only to turbines of relatively large capacity but also to small ones.

The 50,000-kw turbine at Västerås, Sweden, has been brought up to full power from a cold state within 9 min. This starting time includes all the necessary provisions for operation such as starting of all auxiliary machines, draining of the steam system and obtaining full vacuum. This turbine can be brought up to full-speed operation in only two minutes from the time the required vacuum is attained.

A 37,500-kw unit of this type installed in England is guaranteed to come to full operating speed and load from a standing cold state in 8 min.

As reported by Mr. Gustavsson in *Teknisk Tidskrift* for December 16, 1933, the 20/30,000-kw turbine in the Vartan Power Station of the Stockholm Electricity Works is brought from a cold state to full speed of operation in 2 min without any preliminary preheating but after obtaining the required vacuum.

According to a Swedish manufacturer, this type of turbine may in general be counted upon to pick up full load from a cold state in 5 min, this applying to all designs up to 14,000 kw.

David P. Graham.[3] Mr. Estcourt has prepared a valuable paper that will continue to be useful whenever variable-load possibilities are under consideration. The range in steam output obtained with both gas and oil is remarkable, especially when it is remembered that the original design did not contemplate the possibility of stand-by service.

The method of oil-burner operation used to obtain the extreme burner-capacity range is sometimes known as "constant differential control." One way of accomplishing these results with the type of burner used in the tests is in the manner illustrated in the paper. Another way, which for some plants would be simpler and less expensive, is shown in the accompanying Fig. 1.

A simple differential pressure valve, without stuffing boxes, is located in the oil-supply line to a boiler and the diaphragm is also exposed to the pressure in the oil-return line. As the control-valve position is varied with steam demand the differential valve automatically varies the oil pressure at the burner inlet by a like amount.

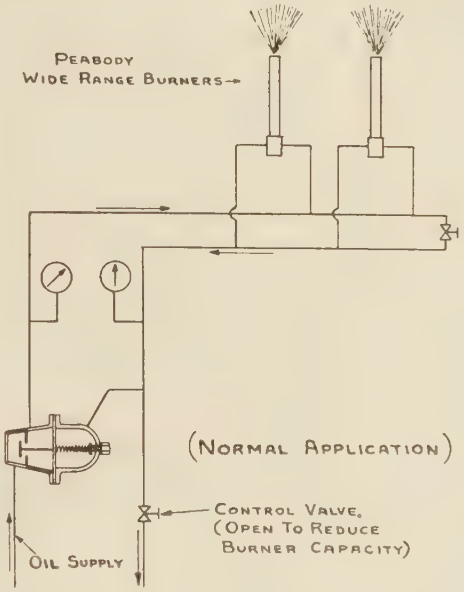

Fig. 1

Such extreme oil-burner capacity ranges are also obtainable with burners of much higher capacity, i.e., 2 to 3 tons of oil per hr. The capacity range becomes of more and more importance as the burner capacity is increased. As an example, one large boiler is fired by several high-capacity oil burners of this type which operate up to 4400 lb of oil per burner per hr during the day and down to 500 lb of oil per burner per hr during the night. The entire range is covered with the same burners, same tips and all parts in service, and by the use of a single control valve.

R. P. Moore.[4] The designers of steam-electric stations should appreciate the data presented in Mr. Estcourt's paper. His experience and the ideas resulting therefrom should be given thorough consideration in any modern design. One or two points deserve further emphasis. Mr. Estcourt reports incidents within his knowledge in which plants designed for peak load have gone on base load and vice versa. These reversals are not limited to his part of the country. Increasing numbers of interconnections between hydro and steam have made similar situations quite common, so that even though the evidence may be conclusive that a steam station is to operate on one type of load or another, the design should be such that extreme range and flexibility are included.

According to Mr. Estcourt's paper he has not experienced any difficulty with boiler "swell." Apparently, in the cases he mentions, this places no limitation on the rate of pickup of load. It is possible, however, that rapid pickup may cause so much swell as to result in serious water carry-over. The minimum level

[1] Published as paper FSP-59-1, by V. F. Estcourt in the February, 1937, issue of the A.S.M.E. Transactions.
[2] Combustion Engineering Co., New York, N. Y.
[3] Peabody Engineering Corporation, New York, N. Y.
[4] Buffalo-Niagara Electric Corp., Buffalo, N. Y. Mem. A.S.M.E.

ADDITION TO TABLE 4 OF PAPER

Plant	Initial boiler-drum pressure, lb per sq in. gage	Maximum capacity, kw	Turbine water rate at 10% max capacity, lb per kwhr	Total boiler storage at normal operating temperature, lb	Capacity from storage for 20% pressure drop per min, kw	Proportion of max turbine capacity from storage per cent
Huntley No. 2	450	160,000	12.5	788,000	118,600	74.0

ADDITION TO TABLE 5 OF PAPER

Plant	Operating pressure at turbine throttle, lb per sq in.	Operating temperature at turbine throttle, F	Maximum installed turbine capacity, kw	Turbine water rate at max capacity, lb per kwhr	Maximum installed —boiler capacity— 1000 lb per hr	Equiv kw	Ratio boiler to turbine capacity, col 7/col 4
Huntley No. 2	425	750	160,000	10.04	2240	224,000	1.40

of "5 inches in the gage glass" which Mr. Estcourt specifies, might have to be reduced in some designs, which, of course, is a compromise between ability to pick up load and the danger of carry-over. This point deserves further study.

It may be of interest to add the figures for C. R. Huntley Station No. 2 to Mr. Escourt's Tables 4 and 5. Charles R. Huntley Station No. 2 is operated by the Buffalo-Niagara Electric Corporation which is part of the Niagara-Hudson System. It contains two 80,000-kw 1800-rpm turbogenerators and four 560,000-lb per hour boilers.

It is to be noted that Huntley No. 2 exceeds Mr. Estcourt's minimum ratio-of-boiler-to-turbine capacity of 1.35. The column headed "Capacity from storage for a 20 per cent pressure drop per minute" apparently takes account of heat stored in the water only, whereas the steel and the setting would add materially to this figure. The figures probably are comparable, however, and in fact the heat absorbed from the steel and the setting tends to be offset by that absorbed by the turbine bleeder-heater system as pointed out by Mr. Estcourt. His system of comparison, as shown by Tables 4 and 5, does not take into account the possible limitation on load pickup due to swell.

Author's Closure

With reference to Mr. Graham's description of the method of "constant differential control" for increasing the range of Peabody oil burners, it should perhaps be pointed out that, while it is true a much wider range of oil-burner operation may be obtained by maintaining a constant differential between the supply and return pressures, the best results are not necessarily realized by holding this differential constant over the entire load range. For example, on the installation described by the author, the control was adjusted to give a slightly greater differential at extremely light loads, as shown in Fig. 6 of the paper, because somewhat better combustion (with practically no carbon deposit on the furnace floor) was obtained in this manner. Increasing the differential at maximum loads has also been considered in order to widen the flame-angle and shorten the length of the flame under these extreme conditions, thus reducing flame-impingement on the rear wall when excess air is held at a minimum.

Mr. Moore's observations regarding boiler "swell" are interesting. It is true that this problem is entirely absent in the case of the 1400-lb boilers described in the paper in the tests at plant *A*. However, in the plants referred to in the paper which operate at or below 450 lb per sq in., it was not the author's intention to convey the impression that no difficulties in this connection had been encountered. At these pressures, the problem cannot be overlooked even though careful operation may in some cases be all that is required in order to avoid trouble, The minimum water level of 5 in., specified in item *f* on page 15 of the published paper,[1] was intended to apply specifically to the boilers used in the tests described in connection with plant *B*. It was not offered as a general rule which could be counted upon as satisfactory for other installations.

The boilers operating at 425 lb per sq in. in plant *D* are provided with an additional "dry drum" located directly above the main steam drum. A special separator in the former serves to reduce carry-over to a minimum. However, as shown in Table 5 of the paper, column 8, both plants *C* and *D*, at present have so much excess boiler capacity that the demand upon the boilers is relatively small even in the event of a full-load pickup on the turbine, thus minimizing the difficulties from "swell" and carry-over.

However, contemplated future additions to capacity will materially reduce this excessive reserve boiler capacity. Even under the present favorable conditions of plant *D*, carry-over is not entirely absent during sudden demands for full load on the turbine. Plant *C*, on the other hand, has for various reasons only operated on a restricted load-pickup schedule; therefore, no data are available as to its performance during a sudden demand for full load.

These facts serve to bring out one of the advantages of higher operating pressures for the type of stand-by operation discussed in the paper. Actual operating experience with both low- and high-pressure boilers leads to the conclusion that the choice of pressures for any contemplated new installation, intended for stand-by operation during all or part of the year, should be seriously influenced by the knowledge of the operating advantages known to exist in favor of the higher pressure.

Further Studies of Three-Dimensional Pipe Bends

By WILLIAM HOVGAARD,[1] BROOKLYN, N. Y.

This paper discusses in some detail the secondary terms which, as explained in a previous paper[2] by the author, occur in the expressions for the rotations in pipe bends under the action of couples normal to their plane. Revised formulas for the reaction couples and forces at the anchorages are given, comprising all the secondary terms throughout. The numerical results, when these formulas are applied, are recorded. The effect of omitting all secondary terms throughout the calculation is considered at length.

THIS PAPER discusses in some detail the secondary terms which, as the author explained in a previous paper,[2] occur in the expressions for the rotations in pipe bends under the action of couples normal to their plane. For the sake of easy reference, Fig. 6 of the previous paper is reproduced in this present paper as Fig. 1. The nomenclature used in this paper is the same as that used in the previous paper.[2]

Secondary Rotations

It was explained in cases II and III of the previous paper[2] that when a quarter bend, which is fixed at one end O and free at the other end A (as shown in Figs. 3 and 4 of the previous paper[2]), is subjected to a constant couple M normal to the plane of the bend, there will be produced at A two rotations, a primary rotation, in the same direction as the couple, expressed as

$$\Delta\varphi = 1.806 \frac{MR}{EI} \qquad [1]$$

and a secondary rotation, at right angles to the primary, expressed as

$$\Delta\varphi = 0.150 \frac{MR}{EI} \qquad [2]$$

The ratio existing between these rotations is as 12 is to 1.

When the couple is variable, as in cases I and IV of the previous paper,[2] where a force is acting at one end of the bend normal to its plane, there will be produced similar secondary rotations, but this fact was not explained in the previous paper.[2] In order to bring this out, consider case IV, as shown in Fig. 2, where a force F acts upward in the Z-direction at the origin O. Instead of starting, as in the previous paper,[2] from the bending and twisting couples M_b and M_t, we first write the couples

$$\left.\begin{aligned} M_x &= -FR\sin\psi \\ M_y &= FR(1-\cos\psi) \end{aligned}\right\} \qquad [3]$$

Dealing first with M_x, and resolving along the tangent and normal at any point S, we have

$$\left.\begin{aligned} M_{xb} &= -FR\sin\psi\cos\psi \\ M_{xt} &= -FR\sin\psi^2 \end{aligned}\right\} \qquad [4]$$

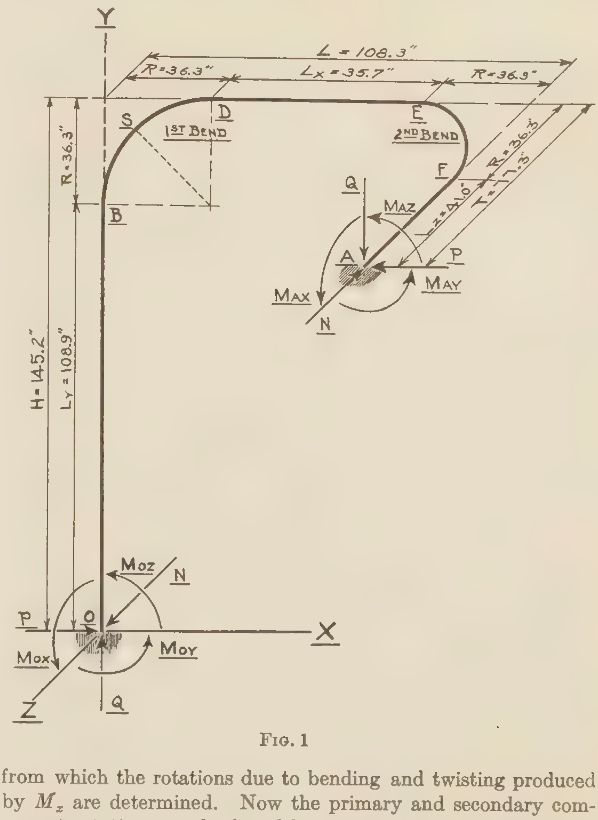

Fig. 1

from which the rotations due to bending and twisting produced by M_x are determined. Now the primary and secondary component rotations can be found by again resolving in the X- and Y-directions. Integrating for the whole bend, the primary or X-rotation at A is

$$\Delta\varphi_{xA} = -\frac{FR^2}{EI}\int_0^{\frac{\pi}{2}} (\sin\psi\cos^2\psi + 1.3\sin\psi^3)\, d\psi = -1.2\frac{FR^2}{EI} \qquad [5]$$

[1] Professor Emeritus, Massachusetts Institute of Technology. Mem. A.S.M.E. Professor Hovgaard was graduated from the Danish Naval College in 1879 with the rank of sublieutenant and from the Royal Naval College, Eng., in 1886. From 1888 to 1900 he served at the Royal Dockyard, Copenhagen, Denmark, alternating with sea duty as a naval officer except for a period between 1895 and 1897 when he was manager of the Burmeister and Wain shipyard, Copenhagen, in charge of design and construction of the Imperial Russian yacht *Standart* and other vessels. From 1901 until 1933 Professor Hovgaard was in charge of the course of Naval Construction at Massachusetts Institute of Technology. He was in the technical service of the U. S. Bureau of Construction and Repair in 1917 and 1918, and served as consultant in the same Bureau from 1919 to 1926. Professor Hovgaard also served on government committees concerning the Navy airship *Shenandoah* and the Army airship RS-1.

[2] "Stresses in Three-Dimensional Pipe Bends," by William Hovgaard, Trans. A.S.M.E., vol. 57, 1935, paper FSP-57-12, pp. 401–416.

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1938, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TABLE 1 COEFFICIENTS FOR PRIMARY AND SECONDARY ROTATIONS AND DISPLACEMENT IN QUARTER BENDS[a]

	Case I			Case II		Case III		Case IV		
	$M_x = FR$ $(1 - \sin\psi)$	$M_y = -FR$ $\sin\psi$	Total	$M_z =$ const	Total	$M_y =$ const	Total	$M_x = -FR$ $\sin\psi$	$M_y = FR$ $(1 - \cos\psi)$	Total
$\Delta\varphi_x$	+0.606	−0.100	+0.506	+1.806	+1.806	+0.150	+0.150	−1.200	+0.050	−1.150
$\Delta\varphi_y$	+0.050	−1.200	−0.150	+0.150	+0.150	+1.806	+1.806	−0.100	+0.606	+0.506
Δz_x	+0.368	+0.041	+1.330	+0.606		+0.050		−0.238	+0.009	−0.476
			−0.082		+0.506		−1.150			+0.068
Δz_y	−0.041	+0.962	+1.248	−0.100		−1.200		+0.059	−0.238	−0.408

[a] Case numbers in this table refer to cases given in the author's previous paper.[2]

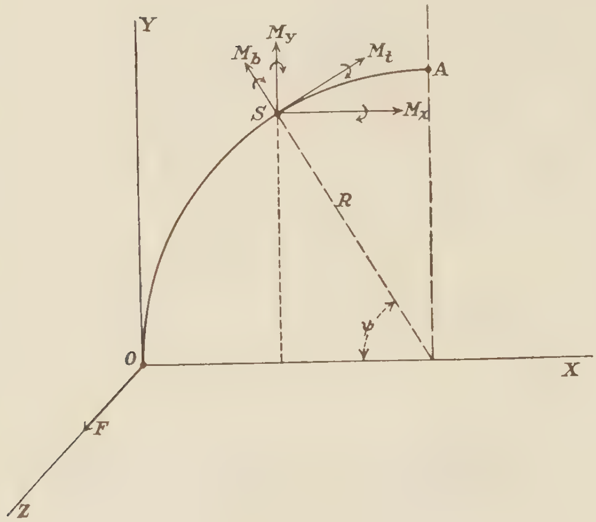

FIG. 2

and the secondary or Y-rotation is

$$\Delta\varphi_{yA} = \frac{FR^2}{EI}\int_0^{\frac{\pi}{2}} (\sin^2\psi\cos\psi - 1.3\sin^2\psi\cos\psi)d\psi = -0.1\frac{FR^2}{EI} \quad \ldots [6]$$

Proceeding in the same manner for M_y, the primary or Y-rotation is

$$\Delta\varphi_{yA} = +0.606\frac{FR^2}{EI} \quad \ldots\ldots\ldots\ldots [7]$$

and the secondary or X-rotation is

$$\Delta\varphi_{xA} = +0.050\frac{FR^2}{EI} \quad \ldots\ldots\ldots\ldots [8]$$

It is seen that also in this case the primary and secondary rotations bear to each other the same ratio of 12 to 1 as in case of the constant couples.

For each one of the four types of rotation previously determined there is a corresponding displacement Δz_A normal to the bend.

Due to the primary rotation caused by M_x we have the displacement

$$\Delta z_{xA} = -\frac{FR^3}{EI}\int_0^{\frac{\pi}{2}} (1 - \sin\psi)(\sin\psi\cos^2\psi + 1.3\sin^3\psi)d\psi$$

$$= -0.238\frac{FR^3}{EI} \quad \ldots [9]$$

while the secondary or Y-rotation caused by M_x produces a displacement

$$\Delta z_{yA} = -\frac{FR^3}{EI}\int_0^{\frac{\pi}{2}} \cos\psi(\sin^2\psi\cos\psi - 1.3\sin^2\psi\cos\psi)d\psi$$

$$= +0.059\frac{FR^3}{EI} \quad \ldots [10]$$

Similarly it is found that M_y produces a primary displacement

$$\Delta z_{yA} = -0.238\frac{FR^3}{EI} \quad \ldots\ldots\ldots\ldots [11]$$

and a secondary displacement

$$\Delta z_{xA} = +0.009\frac{FR^3}{EI} \quad \ldots\ldots\ldots\ldots [12]$$

The rotations and displacements are summarized in Table 1, giving the values of the coefficients not only for case IV but also for cases I, II, and III. It is easy in this table to distinguish the secondary terms, which, except for the displacements, are equal to one twelfth of the primary terms.

In the previous paper,[2] the secondary terms due to the constant couples, where the coefficient is 0.150, were omitted in the rotations, while the secondary terms due to the variable couples were included. In the displacements all secondaries, whatever their origin, were included. It was proposed eventually to correct for this approximation by a recalculation. This method simplified the algebraic solutions and the computations somewhat, but it is admitted that it was somewhat arbitrary and that it leaves a feeling of uncertainty.

Now, it has been found by further study and several numerical calculations, that the increased computations, by including all secondaries for the rotations and the displacements in Equations [25] to [30], inclusive, of the previous paper,[2] is much smaller than first believed; therefore, it seems better to use the complete equations rather than a mixed procedure. These equations are given in the following; the omission of all secondaries will be dealt with separately.

Revised Formulas

The Equations here given include not only all the secondaries and all the corrections explained in the author's closure in the discussion[3] of the previous paper,[2] but in addition the following integral in the X-displacement, which was inadvertently omitted in the paper

$$+\int_R^{R+L_x} T(M_{Oy} + Nx)dx = T(L_xM_{Oy} + \frac{1}{2}L_x^2N + RL_xN) \quad \ldots [13]$$

Complete Formulas for Rotations and Displacements. X-rotation, Equation [25] of the previous paper[2]

[3] Discussion of the paper "Stresses in Three-Dimensional Pipe Bends," by William Hovgaard, Trans. A.S.M.E., vol. 58, July, 1936, pp. 391–400.

$$(3.612\ R + 1.30\ L_x + L_y + L_z)M_{Ox} + 0.150\ RM_{Oy} + 0.150\ RM_{Oz} + 0.150\ RHP + [\tfrac{1}{2}T^2 - (0.150 - 0.006)R^2 - 0.150\ RL_x]Q - [\tfrac{1}{2}L_y^2 + 1.806(L_y + H)R + 1.150\ R^2 + (1.30\ L_x + L_z)H]N = 0 \quad \text{[14]}$$

Y-rotation, Equation [26] of the previous paper[2]

$$+ 0.150\ RM_{Ox} + (1.806\ R + 1.571\ KR + L_x + 1.30\ L_y + L_z) M_{Oy} - (0.571\ KR^2 + \tfrac{1}{2}T^2 - \tfrac{1}{2}R^2)P + (0.506\ R^2 + 2.571\ KR^2 + RL_x + 1.571\ KRL_x + \tfrac{1}{2}L_x^2 + LL_z - 0.150\ RL_y)N = 0. \quad \text{[15]}$$

Z-rotation, Equation [27] of the previous paper[2]

$$+0.150\ RM_{Ox} + (1.806\ R + 1.571\ KR + L_x + L_y + 1.30\ L_z)M_{Oz} + (KR^2 + 1.571\ KRL_y + 1.806\ RH + \tfrac{1}{2}L_y^2 + HL_x + 1.30\ HL_z)P - (2.956\ R^2 + 0.571\ KR^2 + 2.806\ RL_x + \tfrac{1}{2}L_x^2 + 1.30\ LL_z - 0.150\ RH)N = 0 \quad \text{[16]}$$

X-displacement, Equation [28] of the previous paper[2]

$$+0.150\ RTM_{Ox} + (1.30\ L_yT + 1.806\ RT + 1.571\ KL_zR + TL_x + KR^2 + \tfrac{1}{2}L_z^2)M_{Oy} - (HL_y + 0.571\ KR^2 - \tfrac{1}{2}L_y^2)M_{Oz} - (\tfrac{1}{2}HL_y^2 + 0.571\ KL_yR^2 + 0.430\ KR^3 + \tfrac{1}{6}T^3 + \tfrac{1}{3}R^3 + 0.571\ KL_zR^2 + \tfrac{1}{3}L_y^3 - \tfrac{1}{2}TR^2)P + 0.071\ KR^3Q + (0.506\ R^2T + KL_xR^2 + 2.571\ KL_zR^2 + 1.50\ KR^3 + 1.571\ KL_xL_zR + \tfrac{1}{2}TL_x^2 + RL_xT + \tfrac{1}{2}LL_z^2 - 0.150\ RTL_y)N = EI\,\Delta x_A \quad \text{[17]}$$

Y-displacement, Equation [29] of the previous paper[2]

$$- (TL_y + 3.612\ L_zR + 2.956\ R^2 + 1.30\ TL_x + \tfrac{1}{2}L_z^2)M_{Ox} - 0.150\ RTM_{Oy} + (LL_y + \tfrac{1}{2}LL_x + 1.571\ KRL + 0.506\ R^2 - 0.571\ KR^2 - 0.150\ RL_z)M_{Oz} + (\tfrac{1}{2}LL_y^2 + \tfrac{1}{2}HL_x^2 + HL_xR + 1.571\ KRLL_y + KLR^2 - 0.150\ HRL_z - \tfrac{1}{2}KHR^2 + 0.506\ HR^2 - 0.071\ KR^2L_y)P - (\tfrac{1}{6}L_x^3 + 1.506\ R^2L_x + RL_x^2 + 0.571\ KLR^2 + 0.914\ R^3 + \tfrac{1}{6}L_z^3 + 0.356\ R^2L_z + \tfrac{1}{2}RL_z^2 - 0.150\ RL_xL_z - 0.356\ KR^3)Q + [\tfrac{1}{2}TL_y^2 + 1.806\ TRL_y + 1.150\ R^2 (T + H) + 1.806\ RHL_z + 1.30\ THL_x + \tfrac{1}{2}HL_z^2]\ N = EI\,\Delta y_A \quad \text{[18]}$$

Z-displacement, Equation [30] of the previous paper[2]

$$HL_y + 0.356\ R^2 - \tfrac{1}{2}L_y^2 - 0.150\ L_xR)M_{Ox} - (1.30\ LL_y + 2.956\ R^2 + 2.806\ RL_x + \tfrac{1}{2}L_x^2 + 0.571\ KR^2)\ M_{Oy} + 0.071\ KR^3P - (\tfrac{1}{2}HL_y^2 + 0.914\ R^3 + \tfrac{1}{2}LL_x^2 + 1.506\ R^2L_x + 0.785\ KR^3 + 0.571\ KR^2L_x + 0.656\ L_yR^2 - \tfrac{1}{3}L_x^3 - \tfrac{1}{3}L_y^3 - 0.150\ LL_yR)N = EI\,\Delta z_A \quad \text{[19]}$$

Numerical Application of the Formulas. Applying Equations 14] to [19] inclusive, to the same pipe line used in the previous paper,[2] that is, Fig. 1 of this paper, the equations for the reaction forces and the reaction couples at O become

$$27.5\ M_{Ox} + 5.445\ M_{Oy} + 5.445\ M_{Oz} + 790.5\ P + 2604\ Q - 36{,}810\ N = 0 \quad \text{[20]}$$

$$.445\ M_{Ox} + 414.5\ M_{Oy} - 4053\ P + 18{,}877\ P = 0 \quad \text{[21]}$$

$$.445\ M_{Ox} + 394.2\ M_{Oz} + 45{,}620\ P - 15{,}660\ Q - 790.5\ N = 0 \quad \text{[22]}$$

$$21.0\ M_{Ox} + 27{,}990\ M_{Oy} - 11{,}600\ M_{Oz} - 777{,}850\ P + 7780\ Q + 1{,}027{,}700\ N = 468.4 \times 10^6 \quad \text{[23]}$$

$$-22{,}120\ M_{Ox} - 421.0\ M_{Oy} + 26{,}600\ M_{Oz} + 2{,}611{,}300\ P - 369{,}450\ Q + 2{,}380{,}500\ N = 755.2 \times 10^6 \quad \text{[24]}$$

$$10{,}150\ M_{Ox} - 25{,}220\ M_{Oy} + 7780\ P - 776{,}100\ N = 397.8 \times 10^6 \quad \text{[25]}$$

From Equations [21], [22], and [25] it is easy to obtain expressions for M_{Ox}, M_{Oy}, and M_{Oz} in terms of P, Q, and N, and by substitution in Equations [20], [23], and [24] three equations are obtained from which the forces can be found.

The resulting solution is

P = 1,726.0 lb	M_{Ox} = + 54,715 in-lb
Q = 1,815.0 lb	M_{Oy} = − 12,608 in-lb
N = 631.7 lb	M_{Oz} = −127,162 in-lb

Omission of All Secondary Terms

It is of interest to investigate whether it is permissible entirely to omit all secondary terms, both those that are produced by constant couples and those produced by variable couples, since thereby the mathematical treatment is simplified and an appreciable amount of labor is saved in computation.

An attempt has been made by the author to evaluate the error involved by this approximation, but although so far a complete or quite general solution of this problem has not been obtained, it is believed that some light can be thrown on it.

One line of approach is to study the effect of a small local rotation in a straight pipe line.

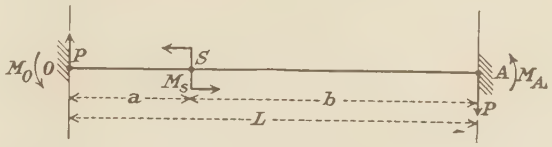

Fig. 3

Consider a straight pipe OA, shown in Fig. 3, of length $L = a + b$. Suppose that a small rotation $\Delta\varphi$ takes place at any point S under the action of a couple M_s, causing reactions at the ends O and A.

If the pipe is hinged at the ends, there will be equal reaction forces $+P$ and $-P$ at O and A, respectively, but no couples, and the point S, where M_s is applied, will move up or down. The reactions P will be inversely proportional to L.

If the pipe is fixed at both ends, as assumed in the present case, there will again be equal reaction forces $+P$ and $-P$ at O and A, respectively, and S will move up or down, but at the same time reaction couples M_O and M_A will be created. By the ordinary method of deflection the following values of these reactions are obtained.

$$P = -\frac{6L}{(a^3 + b^3)} EI\,\Delta\varphi \quad \text{[26]}$$

$$M_O = \frac{(b^2 - ab - 2a^2)L}{a(a^3 + b^3)} EI\,\Delta\varphi \quad \text{[27]}$$

$$M_s = -\frac{L^4}{ab(a^3 + b^3)} EI\,\Delta\varphi \quad \text{[28]}$$

$$M_A = -\frac{(2b^2 + ab - a^2)L}{b(a^3 + b^3)} EI\,\Delta\varphi \quad \text{[29]}$$

It will be seen that for a given rotation $\Delta\varphi$ and for any given ratio between a and b, the reaction forces P are inversely proportional to the square of the length of the pipe, and the terminal reaction couples, as well as the couple M_s at the point of disturbance, are inversely proportional to the length of the pipe to the first power.

The expressions for the couples are of the form

$$M = c\frac{EI\,\Delta\varphi}{L} \qquad [30]$$

where c is a dimensionless coefficient. Fig. 4 gives curves for c and hence for M_O and M_A, for different locations of S. Suppose S to travel from O toward A, then M_O starts at infinity, but falls off rapidly, and at one tenth of L from O, the factor c is about 10. At about one third of L from O, the couple M_O becomes zero and changes sign. Thereafter it remains negative and of moderate amount. At A the value of C is —2.

The curve for M_A is the same as for M_O reversed.

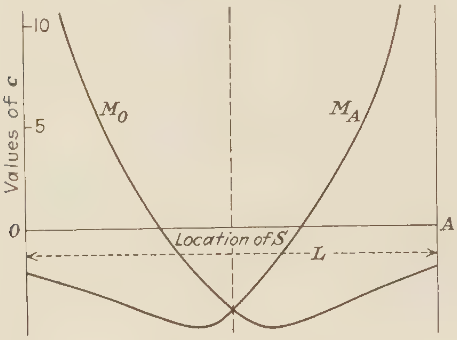

Fig. 4

In practice, secondary rotations are always insignificant close to the anchorages; the maxima are reached at the end of the bends.

It seems obvious from this that in long pipe lines the omission of a small rotation will not in general influence appreciably the calculated reactions at the terminals. If, however, a bend were fitted close to an anchorage in a short pipe line, a small error in the rotation of the end of the bend would cause a relatively great error in the corresponding reaction couple at the nearest anchorage and it is of interest to examine this case more closely.

Referring to the numerical example in the previous paper[2] and using the corrected figures given in this present paper, consider the secondary rotation due to the M_x-couple at the point D in the first bend of Fig. 1. The total secondary rotation due to the combined effect of the constant and variable part of M_x at D (cases II and IV) is

$$\Delta\varphi_y = 0.150\,(M_{Ox} - NL_y)\,\frac{R}{EI} - 0.1\,\frac{NR^2}{EI} = -201 \times 10^{-6} \qquad [31]$$

The omission of this rotation may be regarded as a disturbance, which is imagined to be annulled by a couple M_{Oy}' at O, given by the equation

$$-201 \times 10^{-6} = \frac{M_{Oy}'}{EI}\,(1.806\,R + 1.30\,L_y) \qquad [32]$$

With the values given in the previous paper,[2] we find $M_{Oy}' =$ —773 in-lb; but actually a part of this couple would be taken at A.

In view of the fact that the resultant terminal couple is of the order of 140,000 in-lb, the effect of omitting the secondary rotation is negligible in this case.

The worst condition is where there are no tangents. Applied to the present case this would make $L_y = 0$, and we find $\Delta\varphi_y = 270 \times 10^{-6}$ and then, since the torsion of the straight part L_y is omitted, $M_{Oy}' = 3280$ in-lb, which is still moderate. Actually, the pipe would then consist of two quarter bends only, but the second or EF-quadrant would yield to this rotation with much less effort by bending in its own plane, so that the reaction couple would again be relatively small. This is a special case of the more general one where there are three bends at right angles to one another. Here, the secondary in any one bend will always be largely absorbed as a primary rotation in one of the other bends, which is in the plane of rotation where the flexibility factor K is applied.

In order to throw further light on the problem, complete numerical calculations were made for several pipe lines of the type shown in Fig. 1, comprising two quarter bends of given radius and three tangents, but the length of the tangents was varied. The pipe as given in Fig. 1 represented an intermediate case, and calculations were made for two extreme cases, one where the pipe consisted of only the two quarter bends without tangents and another where the tangents were ten times as long as given in Fig. 1. In all cases one calculation was made in which all secondary terms were included, referred to in the following as the "exact" solution, and another in which all the secondary terms were omitted, referred to as the "approximate" solution.

Table 2 gives the resulting component forces and couples at O as well as the percentage errors by omitting the secondary terms in the three cases. Other intermediate cases were investigated and gave essentially the same results.

The work was carried out largely under the supervision of Henry C. E. Meyer, chief engineer of Gibbs & Cox, Inc., by two of his assistants, Alan Osbourne and Horace L. Bickford.

It is seen from Table 2 that the errors by the approximate method in the cases here studied are on the whole small, bearing in mind that high percentage differences are of importance only where the numerical quantities are great. For instance, the error of 47 per cent, which occurs in the bend with long tangents for M_{Oy}, stands numerically for a difference of only about 200 in-lb in the reaction couple. The errors in the maximum couples in the double quarter bend and in the intermediate case are less than 0.5 per cent. The greatest numerical error is in M_{Ox} for the double quarter bend, where the difference is about 8000 in-lb, but this is only 1.7 per cent of the total resultant reaction couple at O.

On the whole it appears permissible to omit all the secondary terms, but it may be advisable in certain cases to make a rough calculation of the errors involved.

TABLE 2 EFFECT OF OMITTING ALL SECONDARY TERMS

	Double quarter bends, no tangents			Intermediate case, Fig. 1			Long tangents		
	Exact	Approximate	Difference, per cent	Exact	Approximate	Difference, per cent	Exact	Approximate	Difference, per cent
P, lb.	23292	23153	0.60	1726.0	1731	0.3	37	37	0
Q, lb.	9017	8912	1.20	1815.0	1821	0.3	93	92	1.1
N, lb.	8967	8899	0.80	631.7	643	1.8	17	17	0
M_{Ox}, in-lb.	222395	214465	3.60	54715.0	55017	0.5	9940	9342	6.0
M_{Oy}, in-lb.	—186356	—184490	1.00	—12608.0	—13487	7.0	—415	—610	47.0
M_{Oz}, in-lb.	—379747	—378798	0.25	—127162.0	—127717	0.4	—16645	—16358	1.7

Air Chambers for Discharge Pipes[1]

By LORENZO ALLIEVI,[2] ROME, ITALY

The author discusses the application of the general formulas of perturbed flow to the functioning of air chambers connected into discharge pipes. The law of oscillation of pressure rises and falls after the sudden stopping of the power actuating the pump is fully investigated. This law is expressed by an interlocked system of relations, each of which ties together three consecutive values of the pressure at intervals of phase, and are functions of a single parameter representing the ratio of the potential elastic energy of the air chamber and potential elastic energy of the piping.

This pendular system of finite differences is further investigated and modified with a view to giving consideration to the frictional resistance of the liquid, and it is found that (except for small discharges) it is not proper to neglect the influence of such resistance on the damping of the superpressures resulting from the sudden cessation of power, unless the volume of the chamber exceeds practical limits.

Finally, by means of approximate energy balances, extremely simple rules are found for the selection of the dimensions of the chamber and for the increased coefficient of friction of the descending flow.

IN AN earlier study,[3] based on the formulas of perturbed hydraulic motion, the author investigated the problem of the arresting of an ascending liquid column, such as would occur in the discharge pipe of a pump in the case of a sudden stoppage of power. In that study the author examined an arrangement to limit the induced superpressures by means of a flywheel, the kinetic energy of which suffices to secure the safe functioning of the pump during the progressive closure of the influx opening.

Another arrangement, based on the utilization of the accumulated energy—not kinetic, but potential—came recently into use and is met with favor by engineers; this arrangement is the air chamber.

The object of the present study is the rigorous investigation—based on the formulas of perturbed hydraulic motion—of the problem of air chambers, their functioning as related to liquid friction and the determination of their volume, and of the additional frictional resistance required to insure their efficiency.

The problem, so formulated, until now received only a vague and insufficient solution, particularly because the authors of preceding studies systematically neglected the consideration of the potential energy of the pipe line as compared to the analogous energy of the air chamber.

The author will prove that this is not correct and that, to the contrary, the ratio of these two quantities of energy constitutes the parameter of the laws of pressure variations, the key to the problem.

Let us then conceive a plant consisting of a pump of any kind, a suction pipe with a foot valve, and a discharge tube into which is connected an air chamber and which feeds an upper reservoir, the elevation y_0 of which is maintained constant. Let us assume, further, that the arrangement of the efflux from the pipe into the reservoir is such as to also permit, at any time, the flow from the reservoir back into the pipe line. With the sudden stoppage of the power actuating the pump and the coincident closure of the foot valve, there are realized the instantaneous initial conditions of perturbed motion in the system formed by the air chamber, the pipe line, and the reservoir to be fed.

During such instantaneous initial conditions the liquid column still possesses the uniform velocity of regimen V_0, while at its uppermost layer acts the atmospheric pressure h and at its lowest filament the pressure $y_0 + h$, or rather, the pressure $y_0 + h + kv_0^2$ if account be taken of the resistance of liquid friction.

But such instantaneous initial conditions are rapidly disturbed. The ascending liquid column, by inertia, and functioning like a suction piston, generates a drop in pressure at the lower end and causes a flow of water from the air chamber due to the diminished pressure, according to the laws of physics, while the velocity of the column diminishes until it is completely stopped at the instant of maximum drop in pressure. There follows a period of negative (descending) velocity with a return flow of water into the air chamber, accompanied by a rise in pressure to a maximum value $> y_0$ (water hammer) at the instant at which the velocity again becomes zero; this sequence is continued in a rhythmic oscillation the intensity of which is reduced by the influence of liquid friction. The maximum and minimum of velocity and pressure obviously not being synchronous for the various filaments of the liquid column, it is considered sufficient to study the laws applying to the lowest liquid layer in immediate contact with the air chamber.

These studies will be pursued by means of (*a*) the general laws of perturbed motion which tie together, for each layer, the pressures and velocities at intervals of phase (relations of acceleration); and (*b*) the physical laws (isothermic or adiabatic) which tie the pressure and the volume of the air contained in the chamber (volumetric relations).

The author will first treat the problem disregarding liquid friction and will establish a system of pendular relations of finite

[1] Translated from the Italian by E. E. Halmos, Chief Engineer, Parsons, Klapp, Brinckerhoff and Douglas, Neville Building, North Platte, Neb.

[2] Signor Allievi, honorary member A.S.M.E., has been active in engineering work in Italy for many years, but is best known for his basic contributions to the theory of water hammer. In 1903 he published his first notes on "Water-Hammer Theory" and later, in 1913, added to the original treatise. These two works first appeared in Italian and were promptly translated into German and French. In 1925 E. E. Halmos completed the translation into English. These basic contributions are the foundation for practically all of the water-hammer studies which have been made throughout the world since that time and over 100 articles have been published based upon them. Signor Allievi has contributed other studies on water hammer from time to time. It is evident that much of the development and progress in the design of hydroelectric plants, water works, and other types of engineering structures in which liquids are handled has been aided greatly by a knowledge of the basic theory of surges as developed by him.

[3] "L'Elettrotecnica," October 25, 1934.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

differences which rigorously define the laws of pressure at intervals of the phase. He will then introduce in such a system, terms representing the resistance of liquid friction, and find, by a method of corrections, the new values of pressures dependent on such resistance.

Finally, with the help of approximate energy balances, the author will propose rational rules for the selection of values to be assigned, in individual cases, to the volume of the air chamber and to the necessary resistance of liquid friction for descending flow.

General Formulas and Acceleration Relations

Neglecting liquid friction, the general law of perturbed flow is

$$Y_{i-1} + Y_i - 2\, y_0 = (a/g)(\pm V_{i-1} \mp V_i) \ldots\ldots [1]$$

tying the two superpressures $Y_{i-1} - y_0$ and $Y_i - y_0$ at the phase interval $\mu(\mu = 2L/a)$, to the acceleration of the liquid filament during that interval (upper signs for descending, lower signs for ascending motion).

In Equation [1] the atmospheric pressure is assumed to be zero which it obviously is not, particularly because of the arrangement in which an air chamber is inserted; therefore (adopting the lower signs), the formula must be written in the form

$$(Y_{i-1} + h) + (Y_i + h) - 2(y_0 + h) = (a/g)(-V_{i-1} + V_i) . [1a]$$

which, by means of the notations

$$y_0^* = y_0 + h \text{ and } Z_i = Y_i + h/y_0^* \ldots\ldots\ldots\ldots [2]$$

may be written

$$Z_{i-1} + Z_i - 2 = (av_0/gy_0^*)(-V_{i-1} + V_i)/v_0 \ldots . [2a]$$

Remembering, further, the conventional notation $\rho = av_0/2gy_0$, where ρ is the characteristic which defines a pipe line not supplied with an air chamber, and adopting the analogous notation

$$\rho^* = av_0/2gy_0^* \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

Equation [2a] may be written

$$Z_{i-1} + Z_i - 2 = 2\rho^*(-V_{i-1} + V_i)/v_0 \ldots\ldots [4]$$

from which, for $i = 1, 2, 3, \ldots\ldots$ etc., there follows the system

$$\left.\begin{array}{l} -v_0 + v_1 = (v_0/2\rho^*)(Z_1 - 1) \\ -V_1 + V_2 = (v_0/2\rho^*)(Z_1 + Z_2 - 2) \\ -V_2 + V_3 = (v_0/2\rho^*)(Z_2 + Z_3 - 2) \\ \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \end{array}\right\} \ldots\ldots [5]$$

which ties the two series of values v_0, V_1, V_2, V_3, etc., and $Z_0 = 1$, Z_1, Z_2, Z_3, etc., at the instants $t = 0$, μ, 2μ, 3μ, etc., of the total rhythm, beginning at the start of perturbed motion (sudden stoppage of power).

In the following study I shall retain unchanged the first of Equations [5] which will serve, together with the synchronous volumetric equation, to determine the value of the first depression Z_1 which occurs at the end of the phase of the direct blow.

For the subsequent Equations [5] we will substitute new ones obtained by adding each equation to the preceding one; thus we get the new system

$$\left.\begin{array}{l} -v_0 + V_2 = (v_0/2\rho^*)(1 + 2Z_1 + Z_2 - 4) \\ -V_1 + V_3 = (v_0/2\rho^*)(Z_1 + 2Z_2 + Z_3 - 4) \\ -V_2 + V_4 = (v_0/2\rho^*)(Z_2 + 2Z_3 + Z_4 - 4) \\ \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \end{array}\right\} \ldots . [6]$$

which will serve, together with the synchronous volumetric equations, to determine the pressures Z_2, Z_3, Z_4, etc., at the ends of each of the successive phases of the counterblow.

We shall apply the first of Equations [5] and those of Equations [6] to the lowest filament of the liquid in immediate contact with the air chamber; therefore, Z_i shall denote the percentual pressure obtaining there at the ith instant, and $\pm V_i$ shall denote the velocity of the water at that instant flowing out, or entering the air chamber.

Values of ρ^*. As a complement to the preceding discussion, attention is called to the order of magnitude which the characteristic ρ^* may assume within the practically probable limits y_0^* and v_0. Values of ρ^* are given in Table 1. These values increase from ~0.10 to ~2.8 for high or low heads, respectively. Pipe lines having characteristics <~0.10 are excluded, since, with such lines air chambers are nonsensical. The normally found values of ρ^* range from 0.25 to 2.00.

TABLE 1 VALUES OF ρ^*

y_0^*, meters	500	350	200	100	50	25
a, m per sec	1100	1000	900	800	750	700
v_0, m per sec	ρ^* values					
0.5	..	..	0.11	0.20	0.37	0.75
1.0	..	0.14	0.22	0.40	0.75	1.40
1.5	0.16	0.22	0.34	0.60	1.12	2.10
2.0	0.22	0.30	0.45	0.80	1.50	2.80

General Formulas and Volumetric Relations

Designating by C_0 the initial (regimen) volume of air contained in the chamber at the absolute pressure y_0^*, and by C_i its volume at the instant i, then

$$C_i = C_0 + S \int_0^i V\partial t \ldots\ldots\ldots\ldots\ldots [7]$$

where S denotes the constant cross section of the pipe. A second relation between C_0 and C_i is furnished by the physical laws, from which, based on the isothermic assumption $C_0 y_0^* = C_i(Y_i + h)$, which with consideration of Equation [2] gives

$$C_0 = C_i Z_i \ldots\ldots\ldots\ldots\ldots\ldots\ldots [8]$$

Hence, from Equations [7] and [8]

$$\int_0^i V\partial t = \frac{C_0}{S}\left(\frac{1}{Z_i} - 1\right) \ldots\ldots\ldots\ldots [9]$$

From this general volumetric relation it is now possible to arrive at a system of relations, in which, in lieu of the integral $\int V\partial t$, only the V_i of the total rhythm appears.

It may be assumed with reasonable accuracy that during the length of a phase the variation of V can be considered linear and, therefore, it is permissible to make

$$\int_{i-1}^i V\partial t = \mu \frac{V_{i-1} + V_i}{2} \ldots\ldots\ldots\ldots [10]$$

Pressure of Direct Blow. Putting $i = 1$ in Equations [9] and [10], then

$$v_0 + V_1 = \frac{2C_0}{\mu S v_0}\left(\frac{1}{Z_1} - 1\right) v_0 \ldots\ldots\ldots\ldots [11]$$

in which the second member was divided and multiplied by v_0 with a view to the transformation of the coefficient which is given later in this paper.

It is easy to see that Equation [11] combined with the first of Equations [5] permits the elimination of V_1 and results in a second-degree equation for the determination of the pressure Z_1 of the direct blow.

Pressures of the Counterblow. Writing Equation [9] for $(i - 1)$

and $(i + 1)$, adding the equations and subtracting, from the result, Equation [9] multiplied by Equation [2], we obtain

$$-\int_0^{i-1} V\partial t - 2\int_0^{i} V\partial t + \int_0^{i+1} V\partial t + \frac{C_0}{S}\left(-\frac{1}{Z_{i-1}} - \frac{2}{Z_i} + \frac{1}{Z_{i+1}}\right)$$

the first member of which can be written

$$-\int_{i-1}^{i} V\partial t + \int_{i}^{i+1} V\partial t$$

and by Equation [10]

$$-\mu\frac{V_{i-1} + V_i}{2} + \mu\frac{V_i + V_{i+1}}{2} = \mu\frac{-V_{i-1} + V_{i+1}}{2}$$

and the former equation becomes

$$-V_{i-1} + V_{i+1} = \frac{2C_0}{\mu S v_0}\left(\frac{1}{Z_{i-1}} - \frac{2}{Z_i} + \frac{1}{Z_{i+1}}\right)v_0 \quad [12]$$

the second member of which has the same coefficient as Equation [11] and, for $i = 1, 2, 3$, etc., remembering that $Z_0 = 1$, produces a system of equations, each of which tie together three successive values of Z_{i-1}, Z_1, Z_{i+1}, and permits their determination.

Transformation of the Coefficient. $(2C_0/\mu S v_0)$ in Equations [11] and [12]. Prior to performing the indicated eliminations it will be useful to transform the coefficient of the second members of Equations [11] and [12] into an expression which is a function solely of a ratio of energy, so as to confer on the formulas a character of maximum generality.

Let us introduce, therefore, the notation

$$\sigma^* = \frac{1000 C_0 y_0^*}{W_0} \quad [13]$$

in which σ^* expresses the ratio of the potential energy of the air chamber in regimen and the kinetic energy of the discharging liquid column in regimen.

Remembering that in our notation $u = 2L/a$, the coefficient considered can be written

$$\frac{2C_0}{\mu S v_0} = \frac{1000 C_0 Y_0^*}{(1000 L S v_0^2)/(2g)}\,\frac{a v_0}{2 g y_0^*} = \sigma^*\rho^* \quad [14]$$

Therefore, Equation [11] can be written

$$v_0 + V_1 = \sigma^*\rho^*\left(\frac{1}{Z_1} - 1\right)v_0 \quad [15]$$

and Equation [12] may be written

$$-V_{i-1} + V_{i+1} = \sigma^*\rho^*\left(\frac{1}{Z_{i-1}} - \frac{2}{Z_i} + \frac{1}{Z_{i+1}}\right)v_0 \quad [16]$$

the final form of the fundamental volumetric relations for the solution of the problem.

Determination of the Pressures Z_1 of the Direct Blow and $Z_2 \ldots\ldots Z_i$ of the Counterblow

As stated previously, the determination of the pressure Z_i of total rhythm is done by first finding the direct blow Z_1 by means of the first of Equations [5] and Equation [15], and the successive values of the pressure of the counterblow $Z_2 \ldots\ldots Z_i$ by means of Equations [6] and [16] as follows:

Pressure Z_1 of the Direct Blow. Eliminating V_1 from the first of Equations [5] and from Equation [15] and putting

$$N = 2\sigma^*\rho^{*2} \quad [17]$$

there is easily obtained for the determination of Z_1

$$Z_1^2 + Z_1(N + 4\rho^* - 1) - N = 0 \quad [18]$$

an equation which always gives $0 < Z_1 < 1$, and, for a given pipe line (given ρ^*) smaller values for smaller volumes C_0 of the air chamber which is to be expected.

Pressure $Z_2 \ldots\ldots\ldots Z_i$ of the Counterblow. Pendular System. Putting in the general relation given by Equation [16], $i = 1,2,3, \ldots\ldots$etc., and noting that for $i = 1$ there results, $V_{i-1} = v_0$, $Z_{i-1} = Z_0 = 1$, we obtain the system

$$\left.\begin{aligned} -v_0 + V_2 &= \sigma^*\rho^*\left(1 - \frac{2}{Z_1} + \frac{1}{Z_2}\right)v_0 \\ -V_1 + V_3 &= \sigma^*\rho^*\left(\frac{1}{Z_1} - \frac{2}{Z_2} + \frac{1}{Z_3}\right)v_0 \\ -V_2 + V_4 &= \sigma^*\rho^*\left(-\frac{1}{Z_2} - \frac{2}{Z_3} + \frac{1}{Z_4}\right)v_0 \\ &\ldots\ldots\ldots\ldots \end{aligned}\right\} \quad [19]$$

in which the first members are identical with the first members of Equation [6].

Equating, therefore, the second members, and considering Equation [17], we obtain the system

$$\left.\begin{aligned} 1 + 2Z_1 + Z_2 - 4 &= N\left(1 - \frac{2}{Z_1} + \frac{1}{Z_2}\right) \\ Z_1 + 2Z_2 + Z_3 - 4 &= N\left(\frac{1}{Z_1} - \frac{2}{Z_2} + \frac{1}{Z_3}\right) \\ Z_2 + 2Z_3 + Z_4 - 4 &= N\left(\frac{1}{Z_2} - \frac{2}{Z_3} + \frac{1}{Z_4}\right) \\ &\ldots\ldots\ldots\ldots \end{aligned}\right\} \quad [20]$$

which is precisely a pendular system of finite differences, and constitutes, in a sense, a novelty in technical mathematics.

The numerical use of it is obvious: The given pipe line, that is, the given ρ^*, and the selected value of σ^* (which is discussed later in this paper) result, by using Equations [17] and [18], in finding the parameter N and the value of the first depression Z_1, respectively; the first of Equations [20] furnishes then a quadratic equation for determining Z_2, the next analogous equation gives the value of Z_3, etc.

It may be seen that substituting $Z_1 \ldots\ldots Z_i$ into Equation [5] permits the easy determination of V_i, in the form of

$$\left.\begin{aligned} V_1 &= v_0 + \frac{v_0}{2\rho^*}(Z_1 - 1) \\ V_2 &= V_1 + \frac{v_0}{2\rho^*}(Z_1 + Z_2 - 2) \\ V_3 &= V_2 + \frac{v_0}{2\rho^*}(Z_2 + Z_3 - 2) \\ &\ldots\ldots\ldots\ldots \end{aligned}\right\} \quad [5a]$$

and all calculations are extremely easy.

The Parameter N. It will be of interest now to show the energy significance of the parameter

$$N = 2\sigma^*\rho^{*2} \quad [17]$$

which alone appears in the pendular system of Equations [20].

The energy definition of the factor σ^* is given by Equation [13] while regarding the factor ρ^*, we may observe that, disregarding

the atmospheric pressure, the energy expression of the characteristic ρ is

$$\frac{av_0}{2gy_0*} = \rho = {}^1/_2 \sqrt{\frac{W_0}{W}} \ldots\ldots\ldots\ldots\ldots [21]$$

where W designates the elastic potential energy of the pipe line in regimen, considering the atmospheric pressure zero. Therefore, it is logical and justified to formulate the analogous equation.

$$\frac{av_0}{2gy_0*} = \rho^* = {}^1/_2 \sqrt{\frac{W_0}{W^*}} \ldots\ldots\ldots\ldots [21a]$$

where W^* has an analogous meaning to W by the substitution of y_0^* for y_0.

Therefore, by means of Equations [13] and [21a]

$$N = 2\sigma^*\rho^{*2} = \frac{1}{2}\,\frac{1000\ C_0v_0^*}{W^*} \ldots\ldots\ldots\ldots [22]$$

and therefore, the unique parameter of the pendular system of Equations [20] equals the half ratio of the (total) elastic potential energy of the air chamber in regimen, and the analogous elastic potential energy of the discharge pipe, in regimen.

The foregoing statement shows the important significance with which the energical considerations shed light on the investigation of the problem.

Other Observations Regarding the Pendular System. The pendular system Equations [20], independent of v_0 and governed only by the parameter N when the value Z_1 of the pressure of the direct blow is given by Equation [18], is in a certain sense a function of a second parameter, inasmuch as the whole series of successive Z_i depend on the interval of the phase. For instance, if we assume $Z_1 = 1$, Equations [20] are satisfied by $Z_2 = Z_3 = \ldots . Z_i = 1$, the second members of each equation becoming zero (hypothesis of immobility). On the other hand, should Equation [18] furnish a value of $Z_1 < 1$, the system expressed by Equations [20] takes on its pendular character as far as the values of $Z_2 \ldots . Z_i$ are concerned, decreasing over a certain number of rhythms to a value of Z_{min}, then increasing to values > 1 and reaching a value of Z_{max}, after which it again decreases and thus gives a sequence of oscillation.

Analogously, the values V_i (or rather the ratio V_i/v_0) present an oscillation of opposite signs with respect to $\sim Z_{min}$ and $\sim Z_{max}$ and reaching positive or negative maximum values corresponding to $Z \sim 1$. It seems, in fact, unnecessary to observe that values Z_{min} and Z_{max} obtained from Equations [20] are not, in general, the effective minimum or maximum, which obviously occur (as for $Z = 1$) in some instant of intermediate rhythm.

The approximate determination of such effective minimum and maximum could be obtained obviously from the determination of the intermediate values of Z_i.

For a given pipe line (given ρ^*) the duration of a single oscillation is the longer the larger N is and therefore the larger the volume C_0 of the air chamber, which is, of course, logical.

Numerical Examples. To better illustrate the oscillation laws, values of Z_1, Z_{min}, and Z_{max} are given in Table 2 as determined from values of $\rho^* = 0.5$, 1.0, and 2.0, and values of $\sigma^* = 30$, 5.0, and 2.0. These values satisfactorily cover those of actual applications.

TABLE 2 VALUES OF Z, Z_{min}, AND Z_{max} FOR VALUES OF ρ^* AND σ^*

		Z	Z_{min}	Z_{max}
$\rho^* = 0.5$, $\sigma^* =$	30	0.89	$Z_3 = 0.78$	$Z_{10} = 1.30$
	5	0.61	$Z_2 = 0.56$	$Z_5 = 1.71$
	2	0.41	$Z_1 = 0.41$	$Z_4 = 1.93$
$\rho^* = 1.0$, $\sigma^* =$	30	0.94	$Z_1 = 0.78$	$Z_{19} = 1.31$
	5	0.73	$Z_3 = 0.56$	$Z_9 = 1.84$
	2	0.53	$Z_3 = 0.31$	$Z_7 = 3.87$
$\rho^* = 2.0$, $\sigma^* =$	30	0.97	$Z_{14} = 0.78$	$Z_{38} = 1.31$
	5	0.84	$Z_{6,7} = 0.55$	$Z_{16} = 2.53$
	2	0.68	$Z_5 = 0.42$	$Z_{11} = 2.71$

The first observation suggested by the figures in Table 2 is that in a pipe line with a very small air chamber ($\sigma^* = 5$ and 2) there may occur $Z_{max} > 2$ which is larger than the maximum possible value which can demonstrably obtain in a pipe line without an air chamber. With such an assumption, the hydrodynamic phenomenon is ruled, in fact, by the general system of Equations [5], which gives for $V_i = 0$, $Z_1 = 1 - 2\rho^*$, $Z_2 = 1 + 2\rho^*$, $Z_3 = 1 - 2\rho^*$, etc.; these values have a meaning only for the conditions $\rho^* < 0.5$, because Z_1 cannot be <0, from which follows the condition $Z_{max} = Z_2 \gtreqless 2$.

If, on the other hand, $\rho^* > 0.5$, the physical phenomenon obviously is found from Equations [15], and we may easily presume a pressure of counterblow which differs little from 2.

These statements, which are not very favorable to the air chamber, are not paradoxical and indicate only that the hydropneumatic device concentrates the absorption of the energy of the chamber in a very short time with a consequent increase of pressure.

As an example, there is shown in Table 3 the series of the values

TABLE 3 VALUES OF Z_i FOR THE CASE ($\rho^* = 1$, $\sigma^* = 2$)

$Z_1 = 0.531$	$Z_5 = 0.505$
$Z_2 = 0.344$	$Z_6 = 1.369$
$Z_3 = 0.307$	$Z_7 = 3.868$
$Z_4 = 0.341$	$Z_8 = 0.730$

Z_i for the case ($\rho^* = 1$, $\sigma^* = 2$), which gives occasion to a vehement superpressure equal to four times the pressure of regimen. The figures in this table show the whole violence of the phenomenon, concentrated as a blow in the seventh phase.

These statements, therefore, call attention to the study of the dangers of the air chamber and explain many failures of pipe lines due to the neglect of maintaining the chamber, or to lack of air due to defective construction. The author will discuss this subject in more detail later in this paper.

The phenomenon of time of absorption of the energy of the chamber is of a general character and the examination of the series of the values Z_i in Table 2 shows in fact that the duration of the depression is always larger than the duration of the wave of superpressure, while the maximum depression is always less than the maximum superpressure that is $1 - Z_{min} < Z_{max} - 1$.

The two waves, therefore, are nonsymmetrical, and this in proportion to the difference of the two antagonistic forces (gravity and elastic reaction of the chamber) which governs the oscillation.

A continuous curve drawn through the points representing the Z_i of the entire rhythm, therefore, takes the allure of refractory festoons impossible to express analytically.

The pendular system expressed by Equations [20] cannot furnish any criterion or rule for the selection of σ^* or, in other words, for the volume of the air chamber. It is, instead, a perfect instrument of control and furnishes the analytical means for the study of the influence of the resistance of the liquid friction.

Correction of Equations [18] and [20] by Considering Liquid Friction

Assuming, for the resistance of liquid friction in regimen, the well-known expression kv_0^2, in which the coefficient k depends upon L/D and additional resistances; observing, moreover, that the effective pressure at the beginning of the perturbed motion should be rewritten

$$y_0^* = y_0 + h + kv_0^2 \ldots\ldots\ldots\ldots\ldots [23]$$

the numerical values of ρ^* and σ^* should be correlatively corrected.

During the perturbed regimen, moreover, the velocity of each single liquid filament is varying with time, and, at the same instant, is different for each filament. Therefore, the effect of liquid friction is transmitted to the lowest filament which we are considering, according to very complex laws in which it is legitimate to assume that the velocity V_i of the identical lowest filament plays a predominant part. It was considered legitimate, therefore, by various authors on the subject, and the author fully concurs, to adopt kV_i^2 as the corrective term of the pressure Z_i as dependent on the resistance of liquid friction.

Thus, the subtle question arises: How can and should such a correction of our formulas be realized?

First of all it may be admitted that if at a given instant the velocity of the liquid column is positive (that is ascendent) the resistance of liquid friction will tend to hinder the expansion of the air contained in the chamber and therefore increase the pressure, while if the velocity becomes negative (that is descendent), the resistance of the liquid friction will tend to hinder the recompression of the air in the chamber, and therefore, decrease the pressure.

From these considerations we may conclude that, provided the correction is made in a given sense for the period of ascending velocity, it must be made in the opposite sense for the period of descending velocity. An attentive examination of the derivation of Equation [18] (direct blow) and of the pendular system of Equations [20] (counterblow) can furnish more exact criteria for the formulation of the correction sought.

It was stated that the pendular Equations [20] were obtained by equating the second members of Equations [6] and [19], the former expressing the laws of acceleration, and the latter the volumetric laws.

Equations [6]. Equations [6], derived from Equations [5] tie together the acceleration of the filament and two successive superpressures; and, because those parts of the superpressures balancing the liquid friction do not enter in the acceleration of the filament, the general Equation [5] must be corrected to

$$-V_{i-1} + V_i = \frac{v_0}{2\rho^*}\left[Z_{i-1} + Z_i - \left(2 + \frac{kV_{i-1}^2}{y_0^*} + \frac{kV_i^2}{y_0^*}\right)\right]$$

and, therefore, the general system of Equations [6], with the abbreviations

$$k^* = \frac{k}{y_0^*} \text{ and } K_i = k^*V_i^2 \quad \ldots\ldots\ldots\ldots [24]$$

becomes

$$-V_{i-1} + V_{i+1} = \frac{v_0}{2\rho^*}\left[Z_{i-1} + 2Z_i + Z_{i+1} - (4 + K_{i-1} + 2\overline{K}_i + K_{i+1})\right] \ldots [25]$$

Equation [16]. Equation [16], derived from Equations [9], [10], and [13], on the other hand, expresses merely the ratios of the volumes of air and water contained in the chamber, which ratios are not influenced by the liquid friction and have no direct bearing on the values of V_i; therefore, we must maintain Equation [16] without any correction in its second member.

Analogous considerations apply to Equation [18], derived from the first of Equations [5] and Equation [15], whence Equation [18] and the pendular system of Equations [20] corrected with consideration of liquid friction, respectively, must be written

$$Z_1^2 + Z_1(N + 4\rho^* - (1 + K_1) - N = 0 \ldots [18a]$$

$$\left.\begin{aligned} 1 + 2Z_1 + Z_2 - (4 + 2K_1 + K_2) &= N\left(1 - \frac{2}{Z_1} + \frac{1}{Z_2}\right) \\ Z_1 + 2Z_2 + Z_3 - (4 + K_1 + 2K_2 + K_3) &= N\left(\frac{1}{Z_1} - \frac{2}{Z_2} + \frac{1}{Z_3}\right) \\ \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots &\quad \ldots\ldots\ldots\ldots \end{aligned}\right\} \ldots [20a]$$

from which, by determining V_i, we obtain Equations [5] analogously corrected in the form

$$\left.\begin{aligned} V_1 &= v_0 + \frac{v_0}{2\rho^*}\,[Z_1 - (1 + K_1)] \\ V_2 &= V_1 + \frac{v_0}{2\rho^*}\,[Z_1 + Z_2 - (2 + K_1 + K_2)] \\ &\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \end{aligned}\right\} \ldots [5b]$$

observing, for Equations [20a] and [5b], that when V_i results in a negative quantity (descending flow) then K_i must also assume the negative sign. In this case the coefficient k^* may have a larger value than for ascendent flow, depending on the additional resistances, which, as will be seen, we may find opportune to introduce.

The procedure of periodical corrections for the determination of Z_i and V_i by means of Equations [18a], [20a], and [5b] is obvious. Namely for determining Z_1 and V_1 from Equation [18a] and the first of Equations [5b], neglecting K as a first approximation, we obtain approximate values Z_1 and V_1, and from Equations [24] the approximate value K', with which Equation [18a] gives a second approximate value Z_1'' and the first of Equations [5b] a second approximate value V_1'' and hence a value K_1''. In the same manner, a system of values Z_1, V_1, and K_1 are finally obtained which satisfy the two equations and Equation [24] with sufficient accuracy.

Substituting these values in the first of Equations [20a] and the first of Equations [5b], there follows, analogously, the determination of Z_2, V_2, and K_2 which satisfy these equations; we proceed, by means of the second of Equations [20a] and the second of Equations [5b] to the determination of Z_3, V_3, and K_3, etc., and finally, we find a value Z_i, which satisfies

$$Z_{i-1} < Z_i > Z_{i+1}$$

which is the maximum sought.

With some practice, this procedure gives sufficiently speedy results.

It is not superfluous to repeat here a statement regarding Equations [18a] and [20a], already made for Equations [18] and [20], that they do not furnish any direction for the selection of σ^* and, therefore, of C_0.

Numerical Examples. (Dimensions in Meters.) We shall execute the calculation of the series of values Z_i and V_i, for a pipe line defined by $y_0 = 40$, $v_0 = 1$ with the following three assumptions:

(*a*) That the ratio L/D is such that the effect of liquid friction may be neglected. On this assumption, $y_0^* \sim 50$, $\rho^* \sim 1.0$, and taking $\sigma^* = 5$, $N \sim 10$. With these values of ρ^* and N, Equations [18], [20], and [5] give the series of the values Z_i and V_i shown in column *A* of Table 4.

(*b*) That the dimensions of the pipe line satisfy $L/D = 10{,}000$ whence $k \sim 15$, and, with this assumption $y_0^* \sim 65$, $\rho^* \sim 0.77$, while supposing to maintain the same value C_0 gives $\sigma^* = 6.5$, $N = 7.71$, and further $k^* = k/y_0^* = 0.23$. With these values of ρ^*, N and k^*, Equations [18a], [20a], and [5b] give the series of values Z_i, V_i, and K_i shown in column *B* in Table 4.

(*c*) That by means of proper appliances (for example, by fins

TABLE 4 VALUES OF Z_i, V_i, AND K_i

	A		B			C		
i	Z_i	V_i	Z_i	V_i	K_i	Z_i	V_i	K_i
1	0.72	0.86	0.73	0.74	0.127	W	W	W
2	0.60	0.53	0.64	0.25	0.015	W	W	W
3	0.56	0.11	0.64	—0.21	—0.011	0.64	—0.20	—0.037
4	0.57	—0.32	0.79	—0.53	—0.066	0.77	—0.45	—0.184
5	0.65	—0.71	1.09	—0.52	—0.062	0.99	—0.39	—0.143
6	0.83	—0.97	1.52	—0.27	—0.017	1.27	—0.13	—0.015
7	1.21	—0.96	1.59	+0.18	+0.007	1.42	+0.32	+0.023
8	1.82	—0.44	1.18	+0.68	+0.107	1.26	+0.67	+0.105
9	1.84	—0.39	...	...	...	...	...	...
10	1.23	+0.92	. .	...	...	. .	. .	

NOTE: The maximum pressure $Z_9 = 1.838$ given by assumption A reduces to $Z_7 = 1.592$ for assumption B and to $Z_7 = 1.420$ for assumption C.

arranged in the direction of ascending flow between the chamber and the pipe line) the coefficient k^* increases four times for descending flow ($k^* = 4 \times 0.23 = 0.92$); we then obtain a series of values of Z_i, V_i, and K_i, shown in column C of Table 4.

Approximate Values of σ^* From Energy Balance

Approximate values of σ^* may be derived from energy balances approximate to the extent, implying that such balances are an expression of the kinetic energy in any instant, that it is assumed that the same variable velocity applies in each instant to all liquid filaments. This is the same as assuming the elasticity of both water and pipe line to be zero, hence the condition $a = \infty$ which tears down the whole analytical edifice of the theory of perturbed flow.

This hypothesis creates an unreal mechanical system the only elastic member of which is the air chamber, in a manner that, aside from liquid friction, (a) at the instant of minimum pressure one has $V = 0$ and V changes sign; (b) at the instant when the pressure takes the value $Z_0 = 1$ we have $V = —v_0$; and (c) at the instant when the pressure reaches its maximum value, we have again $V = 0$.

Energy balances can therefore be established at the instants (a) and (b) for the determination of $Z_{\min}$, or for the instants (b) and (c) for the determination of $Z_{\max}$, which latter is the one of real interest and which we will designate by Z_m.

Energy balance for ($Z = 1$, $V = v_0$) and ($Z = Z_m$, $V = 0$). The energies considered will, therefore, be for the inelastic hypothesis exclusively.

(α) The kinetic energy W_0 which is dissipated during the interval.

(β) The energy absorbed by the air chamber for isothermic compression for $Z = 1$ to $Z = Z_m$ expressed by ($1000\ C_0 y_0^* Ln\ Z_m$).

(γ) The potential energy of the volume of water $C_0 — C_{\min}$ descending from the upper reservoir into the chamber, is expressed by [$1000\ C_0 y_0^*\ (1 — Z_m^{-1})$]. To the terms ($\beta$) and ($\gamma$) we should add other analogies dependent entirely on the neglect of the elasticity of the pipe line, which when formally observing the hypothesis, we assume to be concentrated in the air chamber:

(δ) The energy ΔW_1^* the increment of the potential energy W_1^*, which can be calculated as follows: We have from Equation [21a]

$$W^* = W_0 \left(\frac{g y_0^*}{a v_0} \right)^2$$

hence, at the instant of maximum pressure Z_m, we have

$$\Delta W^* = W_0 \frac{Z_m^2 — 1}{(2\rho^*)^2} \ldots\ldots\ldots\ldots\ldots [26]$$

(ϵ) The potential energy $1000\ \Delta T y_0^*$ of the volume of water descending from the reservoir into the pipe line dilated from T to $(T + \Delta T)$ from which ΔT can be calculated by expressing ΔW^* as a product of ΔT and the average pressure $(1/2)\ y_0^*\ (Z_m + 1)$, whence

$$1000\ \Delta T y_0^* = 2W_0 \frac{Z_m — 1}{(2\rho^*)^2} \ldots\ldots\ldots\ldots [27]$$

Forming, therefore the energy balance

$$\alpha = \beta — \gamma + \text{Equation [26]} — \text{Equation [27]}$$

we obtain, after reduction

$$W_0 = 1000\ C_0 y_0^* \left(Ln Z_m + \frac{1}{Z_m} — 1 \right) + W_0 \left(\frac{Z_m — 1}{2\rho^*} \right)^2$$

which, divided by W_0 and putting, according to Equation [13]

$$\sigma^* = 1000\ C_0 y_0^* / W_0$$

may be written

$$\sigma^* = \left[1 — \left(\frac{Z_m — 1}{2\rho^*} \right)^2 \right] \left(Ln Z_m + \frac{1}{Z_m} — 1 \right)^{-1} \ldots [28]$$

being an interesting approximate expression of the energy ratio σ^*.

This expression is the product of two factors, the first of which a function of Z_m and ρ^*, represents the influence of the elasticity of the pipe line, and the second the influence of the air chamber. Let us examine them separately.

The first factor results in values close to unity for sufficiently small values of the ratio $(Z_m — 1)2\rho^*$ and on this assumption, the elasticity of the pipe line can be neglected with respect to that of the air chamber, but it diminishes rapidly with increasing values of Z_m and vanishes for $Z_m = 1 + 2\rho^*$ with which assumption $\sigma^* = 0$, $C_0 = 0$.

In fact, it is obvious that without the air chamber the maximum pressure is the first pressure of the counterblow $Z_2 = 1 + 2\rho^*$, as found from the first two of Equations [5] with $V_1 = V_2 = 0$. On this assumption the elastic dilation of the pipe line absorbs the whole kinetic energy W_0.

The second factor (influence of the air chamber) results in ∞ for $Z_m = 1$ which is obvious, but decreases rapidly with increasing Z_m without ever vanishing.

With the data of the discussion which follows (see section on the Coefficient k^* of Liquid Friction for Descending Flow), there is formed a table of the values of σ^* from Equation [28], for values of ρ^* from 0.20 to 2.5 and for the values of the superpressures $Z_m — 1$ from 0.30 to 4.

TABLE 5 VALUE OF σ^* FROM EQUATION [28]

$Z_m — 1 =$	0.30	0.40	0.50	0.75	1.0	1.5	2.0	3.0	4.0
$\rho^* = 0.20$	13.6	0.0	...	...	...	...	...	...	...
$= 0.25$	20.0	12.6	0.0	...	...	...	...	...	...
$= 0.375$	26.7	14.1	7.02	3.00	...	...	...	...	...
$= 0.50$	28.8	16.6	10.4	3.33	0.00	...	...	...	...
$= 0.75$	30.5	18.3	12.3	5.72	2.88	0.00	...	...	..
$= 1.00$	31.0	18.9	13.0	6.55	3.88	1.38	0.00	...	..
$= 1.50$	31.4	19.4	13.5	7.15	4.60	2.37	1.28	0.00	...
$= 2.00$	31.5	19.5	13.7	7.36	4.86	2.71	1.73	0.69	0.00
$= 2.50$	31.6	19.6	13.8	7.46	4.97	2.87	1.94	1.00	0.45

Grade of Approximations of Equation [28]. The assumption that all liquid filaments move with equal velocity, which is necessary for the setting up of the energy balances, deviates sufficiently from the physical reality to have us expect that the effective maximum pressure is greater than the maximum hypothetical pressure obtained on the basis of Equation [28].

This, however, is true only in a slight measure when the hydrodynamic phenomenon is but little accentuated, as, for instance, when $Z_m = 1.3$, $\sigma^* \sim 30$, while it is already noticeable for $Z_m =$

1.5, which statement may be proved by means of Equations [18] and [20].

For uniformity of treatment let us assume in the following three examples $Z_m = 1 + \rho^*$, and, $\rho^* = 0.5, 1.0, 2.0$, respectively.

Example 1. Let $\rho^* = 0.5$, $Z_m = 1.5$, and $\sigma^* = 10.4$, while from Equations [18] and [20], with $N = 5.2$, we have

$$Z_1 = 0.739 \qquad Z_3 = 0.677 \qquad Z_5 = 1.211$$
$$Z_2 = 0.650 \qquad Z_4 = 0.888 \qquad Z_0 = 1.982$$

that is, a maximum effective pressure certainly > 1.982 instead of $Z_m = 1.5$.

Example 2. Let $\rho^* = 1.0$, $Z_m = 2.0$, and $\sigma^* = 3.88$, while from Equations [18] and [20], with $N = 7.8$, we have

$$Z_1 = 0.678 \qquad Z_3 = 0.520 \qquad Z_5 = 0.835 \qquad Z_7 = 2.581$$
$$Z_2 = 0.553 \qquad Z_4 = 0.605 \qquad Z_6 = 1.707$$

that is, a maximum effective pressure certainly > 2.581 instead of $Z_m = 2$

Example 3. Let $\rho^* = 2.0$, $Z_m = 3.0$, and $\sigma^* = 1.73$, while from Equations [18] and [20], with $N = 13.8$, we have

$$Z_1 = 0.645 \qquad Z_5 = 0.400 \qquad Z_9 = 0.912$$
$$Z_2 = 0.496 \qquad Z_6 = 0.430 \qquad Z_{10} = 1.646$$
$$Z_3 = 0.427 \qquad Z_7 = 0.502 \qquad Z_{11} = 2.934$$
$$Z_4 = 0.399 \qquad Z_8 = 0.628 \qquad Z_{12} = 2.350$$

hence it appears that between Z_{11} and Z_{12} there is a maximum considerably > 3.

These statements intend to throw light on the mediocre approximations of the inelastic hypothesis (although corrected by the first factor of the second member of Equation [28]) but not to detract from the value and importance of Equation [28], the only, if somewhat vague, explicit relation by which it is possible to establish in an approximate manner, the value of the maximum pressure and the volume of the air chamber.

Determination of C_0 by Means of the Energy Relation σ^* and Neglecting Liquid Friction

From Equation [13] of σ^* we have

$$C_0 = \frac{W_0\sigma^*}{1000\ y_0^*}$$

from which, by substitution of

$$W_0 = 1000\ LS\frac{v_0^2}{2g}$$

we obtain for C_0 an expression which, obviously may be written

$$C_0 = \frac{LSv_0}{a}\ \frac{av_0}{2gy_0^*}\sigma^* = \frac{LQ}{a}\ \rho^*\sigma^*$$

and assuming, for the purpose of our generic and explorative studies, $a = 1000$ meters, then

$$\frac{C_0}{LQ_0} \sim 0.001\ \rho^*\sigma^* \qquad [29]$$

showing that C_0 is proportional to the length of the pipe line, to the volume of flow in regimen and to the product of the energy ratios ρ^* and σ^*.

Let us examine these several elements. The length L and the normal discharge Q_0 are given in each individual case, and so is the characteristic ρ^* when the normal velocity v_0 is fixed. For the energy ratio σ^*, defined by Equation [28] and essentially depending on the maximum pressure Z_m, we cannot always assign precise rules, the pressure Z_m being, in each individual case necessarily proportional to the head y_0 and to the resistance of the pipe line.

For low heads and small discharges the pipe lines frequently present limits of resistance which admit of superpressures equal to or even multiples of the pressure in regimen, while such is not plausible for high or even medium heads and discharges of substantial volume.

Therefore, neglecting special causes which admittedly are usually outside the limit of security given by $Z_m = 1.3$, from Table 5 we have $\sigma^* \sim 30$, whatever the value of ρ^*, for any pipe line.

TABLE 6 VALUE OF C_0, CuM

Q_0 (m³/sec) =		0.025	0.100	0.200	0.500	1.000
L(m) =		200	400	566	895	1265
		C_0, values				
σ = 30	ρ^* = 0.5	0.075	0.60	1.70	6.71	19.00
	= 1.0	0.150	1.20	3.40	13.40	38.00
	= 1.5	0.225	1.80	5.10	20.10	57.00
	= 2.0	0.300	2.40	6.80	26.80	76.00
	= 2.5	0.375	3.00	8.50	33.50	95.00

With this assumption, and by means of Equation [29] there is given in Table 6 the volumes C_0 for values of discharge Q, which values are based on the assumptions (1) that the five pipe lines satisfy the condition $L/D = 1000$, whence the liquid friction becomes negligible and L becomes proportional to $\sqrt{Q_0}$, and (2) that with $v_0 = 1.0$ $L = 200, 400, 566, 895, 1265$ meters, respectively, for the five pipe lines.

From Table 6 it becomes evident that plausible values of C_0 are furnished by Equation [29] only for discharges less than about 100 liters and high heads (small values of ρ^*), while for larger discharges and low heads the values of C_0 are increasing, finally becoming exhorbitant beyond any technical and economical convenience.

The theory, therefore, confirms the fact already well-noted empirically, that is, provided the liquid friction is neglected, the air chambers are very inadequate instruments for the intended purpose of rendering innocuous the kinetic energy of the liquid column by the simple means of pneumatic accumulation. Volumes of air under pressure which contain 30 times such energy ($\sigma^* \sim 30$) and require large chambers, scarcely suffice to limit by 30 per cent the superpressure of the water hammer.

But, on the other hand, because of such large volumes of air, the oscillations of the pressure, as may be shown by means of Equations [18] and [20], will result in units of μ in a long duration; hence, it is desirable to increase the volume of the chamber to permit it to receive the volume of water which must be discharged by it between the initial and final instant when $Z = Z_{min}$, $V = 0$.

To prove such long duration of the oscillation, let us examine the indexes of the values Z_{min} and Z_{max} determined by means of Equation [18] and Equations [20] precisely for $\sigma^* = 30$ and for the usual five values of ρ^*. These indexes are given in Table 6, which indexes of Z_{min} give, in units of μ, the duration of discharge of water from the chamber. Referring to Table 6 for $Q_0 = 1.0$ and $L = 1265$ ($\mu = 2''{,}5$) it is easy to see that such discharge is greater than 20 cu m.

The considerations and statements presented (except for small discharges) indicate the following as a unique solution of the problem: To adopt very small values of σ^* (that is of C_0), arranging adequate resistances of liquid friction for descending flow, which are adapted to destroy the large superpressure which follows the limitation of σ^*.

The empirical technical practice has already shown the way in this direction and the author has only proved the theoretical basis of this orientation.

The Coefficient k^* of Liquid Friction for Descending Flow

Referring to the conclusions of the preceding section of this paper concerning the convenience, or rather, necessity, of adopt-

ing small values of σ^*, let us observe, moreover, that on this assumption, the duration of the oscillation becomes much shorter and V changes sign already in the first rhythms. Therefore, it is permissible, and it is convenient for the clearness of the discussion, to neglect the indeterminate liquid friction of the ascending flow (which constitutes a factor of safety) and to consider, for descending flow only, a coefficient k^* to which, by means of additional resistances, we may give any desired value.

Reduced to such terms, the problem, therefore, presents itself as the selection of the two unknown parameters σ^* and k^*, both independent and arbitrary, but subordinated to the condition that their combined influence must limit the maximum effective pressure to a technically plausible value.

We do not possess any medium of analytical research which would permit a rational solution of such problem, and we must turn to numerical applications and their results intended to bring to light some useful and sufficiently simple uniform laws, with good enough approximations in the field of the numerical values considered.

Study for $v_0 = 1$. From the pipe lines included in Table 5, let us consider those for which $Z_m - 1 = \rho^*$, whence the first factor of Equation [28] of σ^* becomes constant, and equals 0.75. Now take the five corresponding to the five values of ρ^* from 0.5 to 2.5 which sufficiently cover the field of practical applications.

These pipe lines have the same characteristics as the five included in Table 6 in which we assumed $Z_m - 1 = 0.3$ and therefore, $\sigma^* \sim 30$ for all five pipe lines, thus obtaining the exhorbitant values of C_0 in Table 6.

For the five pipe lines of Table 8, assuming $Q = 1000$ cu m, and $L = 1265$ m, we have from Equation [29], for

$\rho^* =$	0.5	1.0	1.5	2.0	2.5
$C_0 =$	6.58	4.90	4.49	4.39	4.38

as compared to the values

$C_0 =$	19.0	38.0	57.0	76.0	95.0

of Table 6.

Based on these premises, when $v_0 = 1$, the objective of limiting and making practically plausible the maximum pressure of the counterblow, for the five pipe lines of Table 8, can be obtained by assuming for each, respectively, $k^* = 2\rho^*$.

To prove this assertion, select the three pipe lines defined by $\rho^* = 0.5, 1.0$, and 2.0; whence $k^* = 1, 2$, and 4, and proceed, by means of the Equations [29a] and [5b], to calculate the values Z_i which are compared to the analogous values calculated by the Equations [20] with $k^* = 0$

We thus obtain the values given in Table 9 which confirm the preceding assertion, inasmuch as the maximum pressures $Z_{max} = 1.982$, 2.581, and 2.934, determined by $k^* = 0$ reduce to 1.067, 1.289, and 1.113 without taking account of the factor of safety obtained by neglecting the liquid friction of ascending flow.

Study for $v_0 \gtrless 1$. A summary examination of the pendular system given by Equations [20a] shows that such study is superfluous inasmuch as it contains $v_0 = 1$. In fact, it is evident that in the pendular system given by Equations [20a] in which the variables Z_i and $K_i = k^*V_i^2$ are tied together, the system of values Z_i will be identically maintained as long as the system of value K_i is identically maintained; but from the relations of Equations [5b] there clearly results the fact that all V_i are proportional to V_0, whence the expression

$$K_i = k^*V_i^2$$

will remain invariable if with the assumption $v_0 \lessgtr 1$, in lieu of $k^* = 2\rho^*$, we put

$$k^* = 2\rho^*/v_0^2 = \frac{1}{g y_0^*} \cdot \frac{a}{v_0} \qquad [30]$$

TABLE 7 INDEXES OF Z_{min} AND Z_{max} FOR $\sigma^* = 30$

ρ^*	0.5	1.0	1.5	2.0	2.5
Z_{min}	Z_8	Z_7	Z_{10}	Z_{14}	Z_{17}
Z_{max}	Z_{10}	Z_{19}	Z_{29}	Z_{38}	Z_{47}

TABLE 8 VALUES OF $Z_m - 1$, σ^* AND N FOR VARIOUS VALUES OF ρ^*

ρ^*	0.50	1.00	1.50	2.00	2.50
$Z_m - 1$	0.50	1.00	1.50	2.00	2.50
σ^*	10.40	3.88	2.37	1.73	1.39
N	5.20	7.66	10.66	13.84	17.37

TABLE 9 VALUES OF Z_i FOR VARIOUS VALUES OF ρ^*, k^* AND

$\rho^* = 0.5$			$\rho^* = 1$			$\rho^* = 2$	
$k^* = 0$	$k^* = 1$		$k^* = 0$	$k^* = 2$		$k^* = 0$	$k^* = 4$
Z_i	Z_i		Z_i	Z_i	i	Z_i	Z_i
0.739	0.739	1	0.678	0.678	1	0.645	0.645
0.650	0.650	2	0.533	0.533	2	0.496	0.496
0.677	0.664	3	0.520	0.520	3	0.427	0.427
0.838	0.744	4	0.605	0.581	4	0.399	0.399
1.211	0.854	5	0.835	0.703	5	0.400	0.400
1.982	0.961	6	1.707	0.882	6	0.430	0.424
1.453	1.040	7	2.581	1.104	7	0.502	0.463
...	1.067	8	1.632	1.289	8	0.628	0.509
...	1.041	9	...	1.282	9	0.912	0.561
...	...	...	...	...	10	1.646	0.622
...	...	...	...	...	11	2.934	0.692
...	...	...	...	...	12	2.350	0.770
...	...	...	...	...	13	...	0.855
...	...	...	...	...	14	...	0.940
...	...	...	...	...	15	...	1.020
...	...	...	...	...	16	...	1.803
...	...	...	...	...	17	...	1.113
...	...	...	...	...	18	...	1.104

which is the general expression of the percentual coefficient of friction.

Comparing Equations [30] and [24], we therefore obtain for the proper coefficient of liquid friction

$$k = \frac{1}{g}\frac{a}{v_0} \sim \frac{100}{v_0} \qquad [31]$$

The extension of this or analogous formulas to the pipe lines not included in Table 8 would seem to require analogous numerical studies, but from it, and always within the limits of sufficient approximation, there can be given the proof of an interesting and somewhat paradoxical propriety of the pendular system given by Equations [20a] as given in the following paragraphs.

Auto-Compensation of the Pendular System Given by Equations [20a] With the Respect to k^* and Z_{max}. This propriety consists in the fact that, varying k^* more or less, correlatively (and merely as a hydrodynamic play) the velocity V_i will vary less or more in a manner that the products $k^*V_i^2$ tend to change little from the initial values with the result that Z_{max} varies very little even for large variations of k^*.

It is useless to argue over the familiar phenomenon, but the numerical characteristics of it are surprising. The author has calculated for the pipe line defined by $\rho^* = 1$ the values of the maximum pressure for $k^* = 1, 2, 3$, and 4 and obtained for these values of k^* the values $Z_{max} = 1.435$, 1.289, 1.320, and 1.484, respectively, while the values of V_i, at the 4th, 5th, 6th, 7th instant are

$k^* =$	1	2	3	4
$V_4 =$	—0.385	—0.342	—0.313	—0.292
$V_5 =$	—0.524	—0.414	—0.346	—0.307
$V_6 =$	—0.455	—0.378	—0.264	—0.247
$V_7 =$	—0.232	—0.195	—0.235	—0.122

which figures confirm the foregoing statement.

It is also interesting to prove that when k^* differs from $k^* = 2$ in either direction, the value of Z_{max} always increases, whence the value corresponding to $k^* = 2$ is a minimum.

Therefore, within such limits of the values k^*, there appears an outside action maintaining the small variations in the field of the

values given in Table 5, and it is best to comply with the technical requirements in every case.

(*a*) To assume, for the volumes C_0, values of the order of magnitude corresponding to the values σ^* which are obtained from Equation [28] with $Z_m - 1 = \rho^*$ (as in Table 8).

(*b*) To assume for the coefficient of liquid friction the value given by Equation [31] without further computations.

Moreover, the formulas derived permit the complete study of each special case.

Pipe Lines Which Do Not Need Additional Resistances. This case occurs when the coefficient k given by Equation [31] equals the natural coefficient of friction of the pipe line, and, therefore

$$\frac{1}{g}\frac{a}{v_0} \sim 0.0015\frac{L}{D} \qquad [32]$$

a condition which is not exceptional.

Pipe Lines Which Do Not Need Air Chambers. Such pipe lines are chiefly those with high heads, characterized by $\rho^* < 0.10$ for which the omission of the air chamber gives $Z_{max} = 1 + 2\rho^*$.

The air chamber can also be omitted for pipe lines for which $\rho^* > 0.10$ when the ratio L/D is sufficiently high to generate a friction able to keep Z_{max} within permissible limits.

The author reserves the treatment of the problem of medium heads for a further discussion.

Air Chambers and Valves in Relation to Water Hammer

By R. W. ANGUS,[1] TORONTO, CANADA

The author gives equations which are necessary for the complete analysis of surge conditions in pump-discharge lines when data for the discharge-line conditions are presented graphically. He selects several types of discharge line and presents graphs which indicate water-hammer conditions therein. Some of the types discussed are simple discharge lines from pumps, discharge lines with an air chamber some distance from the pump and also from the reservoir into which the line discharges, discharge lines wherein the water column breaks, and discharge lines wherein water hammer occurs due to the chattering or slamming of valves.

THE nomenclature used in this paper is that recommended by the A.S.M.E. Committee on Water Hammer, except in the case of the pipe-line constant; the author uses ρ for this constant instead of K. The other notations are as follows:

H_0 = static head, measured during steady flow, above the valve or other device that causes the water hammer, ft

H = head at any point on the pipe during water-hammer effect, ft

H_{At} = head at a point A on the pipe t sec after water hammer begins, ft

$H_{B1.2}$ = head at a point B on the pipe 1.2 sec after water hammer begins, ft

h_{At} = ratio H_{At}/H_0

$F[t-(x/a)]$ = the sum of all the direct pressure waves, in feet of head, due to water hammer at the point x and at time $t-(x/a)$. Also written as F

$f[t+(x/a)]$ = the sum of all the reflected pressure waves, in feet of head, due to water hammer at the point x and at time $t+(x/a)$. Also written as f

V_0 = velocity in pipe for steady flow, fps

V = velocity in pipe at any point during water hammer, fps

V_{At} = velocity in pipe at point A and at t sec after water hammer begins, fps

v_{At} = ratio V_{At}/V_0

a = velocity of the pressure wave, fps

x = distance along the pipe from the cause of the water hammer, ft

L, D, e = respectively, length, diameter, and wall thickness of the pipe, ft

y = vertical distance of a specified point above the gate or nozzle, ft

k = bulk modulus of elasticity of the water, lb per sq ft

E = modulus of elasticity of the pipe-wall material, lb per sq ft

t = time, usually reckoned after water hammer begins, sec

ρ = $aV_0/2gH_0$ is Allievi's pipe-line constant

n = speed of pump, rpm

A, B, etc. = points on the pipe, but A is also used for the pipe area, sq ft

B' and B'' = points on the pipe close to, but on opposite sides of B.

The phenomena accompanying the variable motion of fluids in pipes have been the subject of much study and investigation for years and have led to many solutions often inconsistent and unsatisfactory. It is now generally agreed that any change of velocity is accompanied by the motion of a pressure wave which travels along the pipe in both directions to and from its source and continues to move backward and forward till it is damped out by friction. Where the pipe is of uniform diameter and thickness the nature of these waves and the total pressures produced by them may be found readily, thanks to the work of Joukowsky (1)[2] and of Allievi (2), but where the thickness of the pipe varies and where it changes in diameter and has branches, the problem is much more complicated due to the separate systems of waves set up by each of these causes. These several waves combine with each other in a most troublesome way.

It is quite possible to write mathematical expressions covering the case, but where the initial cause of the waves is somewhat irregular these expressions become difficult in the simple pipe and become so involved in systems of the ordinary type that the solution of them is far beyond the patience of the engineer even if he had the skill to work them. Therefore, the answers to the problems have frequently been obtained by empirical formulas, very limited in application and often leading to most erroneous results. For these reasons the argument of this paper is presented and some illustrations of a common nature are given.

The derivation of certain of the fundamental equations has already been given (3, 4) and need not be further referred to, so that these equations will be stated here without proof. The first are the two equations of Allievi (2) for gate operations, as follows:

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

$$H - H_0 = F\left(t - \frac{x}{a}\right) + f\left(t + \frac{x}{a}\right) \dots\dots\dots [1]$$

$$V_0 - V = \frac{g}{a}\left\{F\left(t - \frac{x}{a}\right) - f\left(t + \frac{x}{a}\right)\right\} \dots\dots [2]$$

where a is the velocity of the pressure wave along the pipe. The symbols used, except where noted otherwise, are those recommended by the A.S.M.E. Water Hammer Committee. Stated in words the first equation says that the rise in pressure $(H - H_0)$ ft above the initial pressure H_0 ft, at any instant t sec after gate movement begins and at a point x ft away from the control gate or other cause of disturbance, is the sum of all the direct or F waves at $[t - (x/a)]$ sec, plus the sum of all the reflected or f waves at $[t + (x/a)]$ sec; as all the reflected waves have an opposite sign to the direct ones, the right-hand side of Equation [1] is a numerical difference while the same side of Equation [2] is an arithmetical sum. These equations apply either to the opening or closing movement of a gate at the end of a pipe but do not include friction or velocity-head effect.

Where the change of velocity is due to a power failure on a pump delivering water through a pipe line to a reservoir, Equations [1] and [2] must be written with a negative sign applied to the left-hand term in each case, otherwise there is no change in the equations, and in this case x is the distance from the pump to the point under consideration, since it is the pump that causes the change of velocity.

By the addition of Equations [1] and [2] and then by subtracting Equation [2] from Equation [1] there result

$$H - H_0 = -\frac{a}{g}(V_0 - V) + 2F\left(t - \frac{x}{a}\right) \dots\dots [3]$$

and

$$H - H_0 = +\frac{a}{g}(V_0 - V) + 2f\left(t + \frac{x}{a}\right) \dots\dots [4]$$

The former of these equations evidently applies to the direct wave as it includes only the F term, while the latter, including as it does only the f term, applies to the reflected wave.

Consider now a pipe ABC, shown in Fig. 1, of diameter D ft

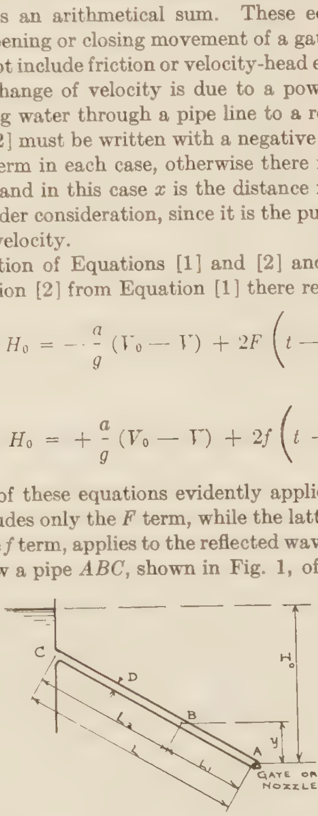

FIG. 1 PIPE LINE WITH CONTROL VALVE AT A

and length L ft and divided at B into lengths L_1 and L_2, the control valve being located at A. Then the time taken for the pressure wave to travel from A to B or B to A is L_1/a sec and to travel from B to C or vice versa requires L_2/a sec, where the wave velocity

$$a = \frac{4665}{\sqrt{\left(1 + \frac{kD}{Ee}\right)}} \dots\dots\dots\dots [5]$$

In this formula D is the diameter of the pipe, in feet; and e is the wall thickness, in feet; while k and E are respectively the bulk modulus of elasticity of the water and the modulus of elasticity of the pipe-wall material, both in the same units (4, 5).

If any pressure wave is started at A at t sec due to an opening or closing of the valve this wave will travel up the pipe with undiminished intensity reaching B at t_1 sec where $t_1 = [t + (L_1/a)]$ and will reach C at t_2 sec, where $t_2 = [t_1 + (L_2/a)] = [t + (L/a)]$ sec. This same wave with undiminished intensity will be reflected from the reservoir and will reach B at t_3 sec, where $t_3 = [t_2 + (L_2/a)]$ sec, and will reach A at t_4 sec, where $t_4 = [t_3 + (L_1/a)] = [t_2 + (L/a)]$ sec.

Let it be supposed that friction and velocity head are at present being neglected, then it is clear that the hydraulic gradient is horizontal, and if the subscript zero is used to denote the condition of steady flow, it follows that

$$H_{A_0} = H_{B_0} = H_{C_0} = H_0 \dots\dots\dots\dots [6]$$

or the heads at A, B, and C are the same at zero time.

Applying Equations [3] and [4] to the pipe gives the following series: (1) For the direct wave

$$H_{At} - H_{A0} = -\frac{a}{g}(V_{A_0} - V_{At}) + 2F(t)$$

$$H_{Bt1} - H_{B0} = -\frac{a}{g}(V_{B0} - V_{Bt1}) + 2F\left(t_1 - \frac{L_1}{a}\right)$$

$$= -\frac{a}{g}(V_{B0} - V_{Bt1}) + 2F(t)$$

$$H_{Ct2} - H_{C0} = -\frac{a}{g}(V_{C0} - V_{Ct2}) + 2F(t)$$

(2) Similarly for the reflected wave

$$H_{Ct2} - H_{C0} = +\frac{a}{g}(V_{C0} - V_{Ct2}) + 2f\left(t_2 + \frac{L}{a}\right)$$

$$= +\frac{a}{g}(V_{C0} - V_{Ct2}) + 2f(t_4)$$

$$H_{Bt3} - H_{B0} = +\frac{a}{g}(V_{B0} - V_{Bt3}) + 2f(t_4)$$

$$H_{At4} - H_{A0} = +\frac{a}{g}(V_{A0} - V_{At4}) + 2f(t_4)$$

Carrying out two subtractions in the direct-wave group gives

$$H_{At} - H_{Bt1} = +\frac{a}{g}(V_{At} - V_{Bt1}) \dots\dots\dots [7]$$

$$H_{Bt1} - H_{Ct2} = +\frac{a}{g}(V_{Bt1} - V_{Ct2}) \dots\dots\dots [8]$$

and for the reflected-wave group

$$H_{Ct2} - H_{Bt3} = -\frac{a}{g}(V_{Ct2} - V_{Bt3}) \dots\dots\dots [9]$$

$$H_{Bt3} - H_{At4} = -\frac{a}{g}(V_{Bt_3} - V_{At4}) \dots\dots\dots [10]$$

On a diagram with coordinates H and V, these four equations are represented by straight lines sloping at angles whose tangents are $+a/g$ and $-a/g$. The equations apply equally well to opening and closing of the gate, i.e., to increase or decrease of velocity and to any method of gate operation as well as to a pipe supplied by a reservoir with stationary or with a variable level and also to pipes supplied at variable pressure. Usually time is reckoned after valve movement begins, in which case $t = 0$. Should there be a change in diameter or thickness at B in such a

way as to change the pressure wave velocity as given by Equation [5] so that there is a wave velocity a_1 in AB and a_2 in BC, the only change necessary in the equations is to use a_1 in Equations [7] and [10] while a_2 would be used in Equations [8] and [9]. Equations [7] to [10] are those employed by Schnyder (6) and Bergeron (7) who developed a graphical method like that given here.

Some engineers prefer to use the ratio of the pressure change H/H_0 instead of the actual rise $H - H_0$, and sometimes this is a more convenient method. If this is desired, Equation [7] may be written

$$\frac{H_{At}}{H_0} - \frac{H_{Bt_1}}{H_0} = 2\frac{aV_0}{2gH_0}\left(\frac{V_{At}}{V_0} - \frac{V_{Bt_1}}{V_0}\right)$$

or more conveniently

$$h_{At} - h_{Bt_1} = 2\rho(v_{At} - v_{Bt_1})\ldots\ldots\ldots\ldots[11]$$

where $h = H/H_0$, $v = V/V_0$, and ρ is Allievi's pipe-line characteristic. The remaining three equations are then

$$h_{Bt_1} - h_{Ct_2} = +2\rho(v_{Bt_1} - v_{Ct_2})\ldots\ldots\ldots[12]$$

$$h_{Ct_2} - h_{Bt_3} = -2\rho(v_{Ct_2} - v_{Bt_3})\ldots\ldots\ldots[13]$$

$$h_{Bt_3} - h_{At_4} = -2\rho(v_{Bt_3} - v_{At_4})\ldots\ldots\ldots[14]$$

The plotting of these equations differs in no way from Equations [7], [8], [9], and [10]; the axes are h and v and the tangents of the slopes are $+2\rho$ and -2ρ. However, some caution must be used, for if the pipe were horizontal, h would be the true ratio of the pressure change in the pipe, but if the pipe were to slope, h would give the ratio referred to H_0, but the ratio of the pressures in the pipe is really $(H - y)/(H_0 - y)$ where y is the elevation of the point on the pipe above the outlet nozzle. See Fig. 1.

Where the pressure changes are caused by stopping a pump, then the direct pressure wave travels in the same direction as the water and not in the opposite direction as is true when a valve at the discharge end is closed. Therefore, for the case of electrical or other power failure on a motor driving a centrifugal pump, the algebraic sign on the right-hand side of Equations [7] to [14], inclusive, would be changed and Equation [7] would therefore be written

$$H_{At} - H_{Bt_1} = -\frac{a}{g}(V_{At} - V_{Bt_1})\ldots\ldots\ldots[15]$$

while Equation [11] would be

$$h_{At} - h_{Bt_1} = -2\rho(v_{At} - v_{Bt_1})\ldots\ldots\ldots[16]$$

and so on with all of the others.

These equations are all that are necessary for the complete analysis of any system, but the positions of the lines they represent cannot be fixed without some further data; however, these

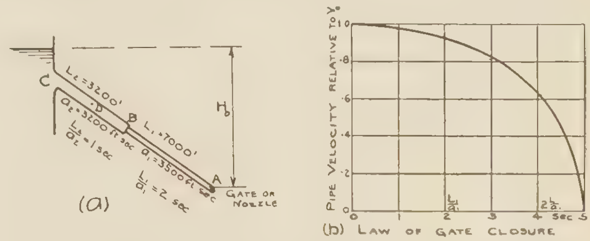

Fig. 2 Pipe Line of Two Diameters (a) and the Curve (b) Representing the Law of Its Gate Closure

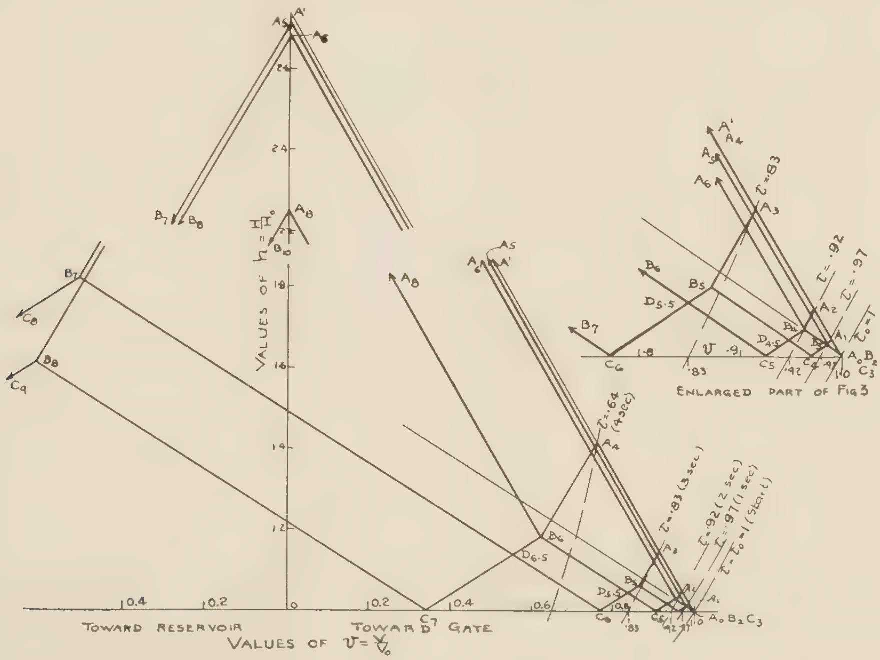

Fig. 3 Effect of Gate Closure According to Fig. 2(b) for the Line Shown in Fig. 2(a)

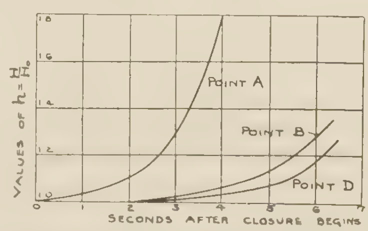

Fig. 4 Pressures for the Case Shown in Figs. 2 and 3

are always available. For example, under steady conditions H_0 and V_0 are always known and $h_0 = 1$ and $v_0 = 1$ so that a point representing the starting conditions is readily placed. Hereafter, time will be reckoned after the movement of the gate or other cause of disturbance begins, so that if a subscript is used after a letter to denote the time at which an event occurs at that point, then A_0 is readily found on the H-V or h-v diagrams and gives the starting point for the plotting.

Usually some other simple facts are available, such as the conditions at the reservoir. If it be large and remain level during the extremely short time of serious water-hammer effects, then all points representing conditions at the reservoir, that is all points C, are fixed on the line $H = H_0$ or $h = h_0 = 1$. It will be advisable, however, to illustrate by some problems.

A pipe of two diameters as shown in Fig. 2(a) draws water from a forebay and discharges through a gate or nozzle of variable area at A. This nozzle is closed in the somewhat irregular way indicated in Fig. 2(b) which represents, on a time base, the pipe velocities corresponding to the various gate positions, times being reckoned after gate closure begins; the discharges are for a constant pressure H_0 on the pipe at the nozzle, and naturally for any other head the corresponding pipe velocity is found by multiplying the velocities on the curve at this nozzle setting by $\sqrt{(H/H_0)} = \sqrt{h}$. For this case the h-v diagram will be used, and hence A_0 is located at $h = 1$, $v = 1$ and, further, for the initial gate opening the pipe velocity ratio v for any other head than H_0 will be represented by a point on a parabola with vertex at $v = 0$, $h = 0$ and which passes through A_0. The reservoir level is assumed fixed.

The plotting is made for the data shown in Fig. 2, and without further information the values of a have been assumed, although for any pipe these may readily be computed by Equation [5]. In Fig. 3 the axes of h and v are shown, also A_0 is readily placed and part of the discharge parabola through it (unnecessary in this case) has been drawn. Since the pressure wave takes 2 sec to reach B and 3 sec to reach C, the points B_2 and C_3 are also located at A_0. But during the first second very little effect is produced on the velocity as is shown by the positions of A_1, A_2, A_3, and A_4 located as shown. The values of τ used on Fig. 3 have been scaled from Fig. 2(b); thus at 2 sec the velocity scales $0.92V_0$ or $\tau = 0.92$. The equations for the case may be written from Equations [11] to [14], inclusive, as follows

$$
\begin{aligned}
h_{A_0} - h_{B_2} &= 2\rho_1(v_{A_0} - v_{B_2})\\
h_{B_2} - h_{C_3} &= 2\rho_2(v_{B_2} - v_{C_3})\\
h_{C_3} - h_{B_4} &= -2\rho_2(v_{C_3} - v_{B_4})\\
h_{B_4} - h_{A_6} &= -2\rho_1(v_{B_4} - v_{A_6})\\
h_{A_2} - h_{B_4} &= 2\rho_1(v_{A_2} - v_{B_4})\\
h_{B_4} - h_{C_5} &= 2\rho_2(v_{B_4} - v_{C_5})
\end{aligned}
$$

and so on throughout the problem. Some of these equations represent an infinitely short line or point, others represent definite lines as is shown in the Fig. 3. Here evidently the maximum pressure at A reaches A_5 which gives nearly the same pressure as sudden closure would have done, as will appear by inspection since closure in $2L_1/a_1$ sec would have given the pressure A'. The pressures and velocities at A, B, and D in the center of BC are plotted in Fig. 4. The problem is solved for $H_0 = 500$ ft and $V_0 = 8$ fps, giving $\rho_1 = 0.87$ and ρ_2 has been taken as 0.35.

In addition to the end points, the conditions at D in the center of BC are marked on Fig. 3 and should require little explanation. Such a point as $D_{5.5}$ is located from the equations

$$h_{C_5} - h_{D_{5.5}} = -2\rho_2(v_{C_5} - v_{D_{5.5}})$$

and

$$h_{B_6} - h_{D_{5.5}} = +\,2\rho_2(v_{B_6} - v_{D_{5.5}})$$

and a similar reasoning applies to other points along the pipes.

Discharge Line From Pump

This type of problem occurs in so many forms that only a few cases can be considered. The first one will be a line from a pump to a reservoir but with an air chamber in the line close to the pump and with a valve between the chamber and pump. This arrangement is shown in Fig. 5 and the problem is similar to one solved by Bergeron (7); it is a very simple application of the theory being discussed.

In this problem it was assumed that the valve was suddenly closed and a study of the pressures in the pipe at the air chamber were made. Full data are given on the drawing and the diagram is plotted with axes of h and v; air changes were assumed isothermal and friction has not been taken into account. The value of $2L/a$ for this pipe is 10 sec, and in such cases as this it is advisable to calculate points at much smaller intervals, the magnitude of which are to be settled by experience; in any event the pressure drop in any interval should not be large, and further it is evident that at the beginning the pressure falls very rapidly and after that at a decreasing rate. Therefore, in this illustration the intervals were one fifth of a second for the first second, then one half second and then one second, as shown in the figure.

The calculation was made in the following way: Starting at the point $h = 1$, $v = 1$, the water-hammer line was drawn below the axis of v with a slope whose tangent is $2\rho = (2 \times 3216 \times 4)/(2 \times 32.16 \times 400) = 1$. In 0.2 sec, taking the pipe area A the demand was $0.2AV = 0.2\,A \times 4$ cu ft, all of which had to come from the air chamber since the valve was closed; hence, the water must have fallen by a distance $(1/5) \times (8/10) = 0.16$ ft because the air chamber has an area five times that of the pipe. Due to the expansion of the air the pressure at C in the air chamber then fell to $(2/2.16) \times 424 = 392.6$ ft abs or to 358.6 ft gage pressure, and the corresponding pressure at A was $358.6 + (10.00 - 0.16) = 368.4$ ft, for which $h = 368.4/400 = 0.921$; this located the point marked 0.2 on the diagram at which the velocity may be scaled as 0.921 corresponding to $4 \times 0.921 = 3.68$ fps.

Proceeding in this way down the same line the final point 10 on it was reached. The next point 10.2 was calculated in the same way, but is on another water-hammer line located as shown, and so on throughout the entire interval of 40 sec for which the calculations were made. The curious shape of the figure on the water-hammer diagram is interesting, and also the shapes of the pressure and velocity curves as shown on the right-hand diagram. While the period of the pressure wave in the pipe is 10 sec, the actual pressure and velocity curves are most irregular and have no definite periods as far as the calculations have been carried.

Pump-Discharge Line With Air Chamber Some Distance From the Pump and Also From the Reservoir Into Which the Line Discharges

This is similar to the former problem, differing from it only with respect to the location of the air chamber. The line considered is shown in Fig. 6, many of the dimensions being marked on this figure; the cross-sectional area of the chamber is four times that of the pipe, $a = 2760$ fps, $V_0 = 5$ fps, and $H_0 = 300$ ft. The air changes in the tank are taken as being isothermal. The air chamber is shown at B' in the center of the line which is 5520 ft long so that L/a for the line is 2 sec, and for each part of it is 1 sec, and the diagram is plotted with axes of H and V for the case of sudden power failure.

From the characteristic curve for the pump at full speed, and from a knowledge of the pump's dimensions, it is possible

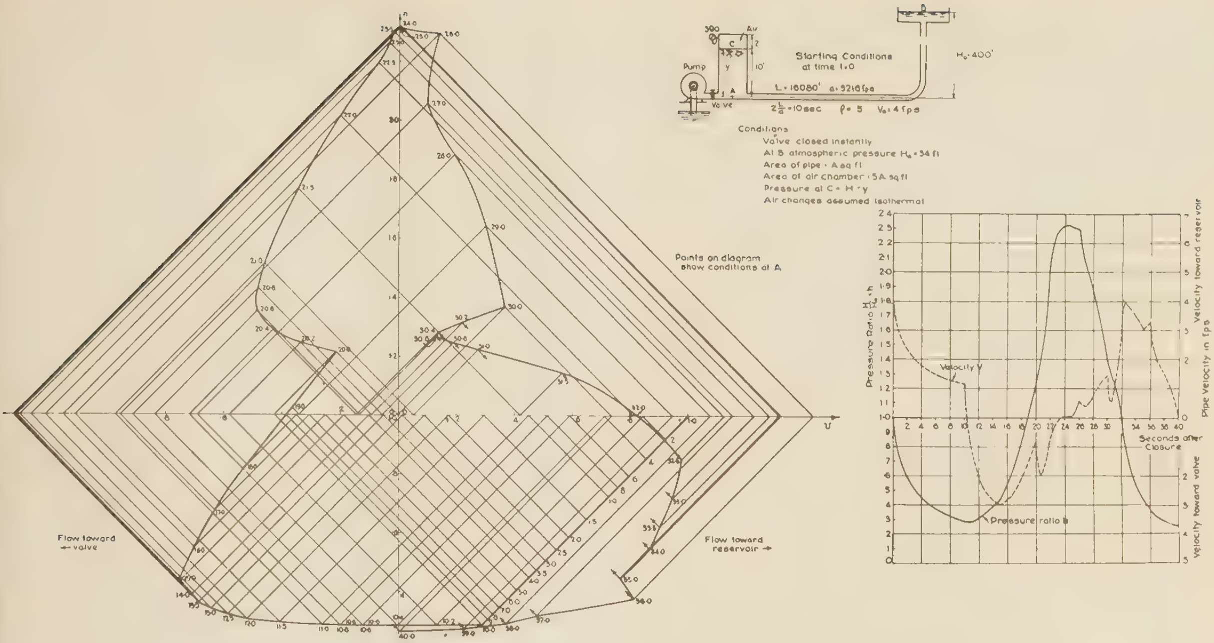

FIG. 5 EFFECT OF SUDDEN VALVE CLOSURE BETWEEN THE PUMP AND AIR CHAMBER SHOWN IN THE SKETCH

to compute the speeds and hence the characteristic curves, for the pump for each 2 sec after power failure, and these have been plotted on the diagram and marked n_2, n_4, and so forth. (This pump has a flywheel.) The water-hammer line is drawn through $A_0B_1C_2$ with slope $a/g = 85.7$, and the points A_2 and A_4 are located; also B_3' coincides with A_2. However, at 3 sec, the pressures at B' and B'' are the same and are known, that is, there has been a fall in pressure from H_0 to H_{B_3}'; therefore, the air in the reservoir will expand and force out some of the water and V_{B_3}'' is greater than V_{B_3}'. The difference between them is easily found, for the air pressure $H_D = H_{B_3}' - Z_1$ from which H_D is known, and hence the following relation may be written: (assuming atmospheric pressure to correspond to 34 ft water column)

$$\frac{H_{B_3}' - Z_1 + 34}{H_0 - Z_0 + 34} = \frac{y_0}{y_1}$$

which gives y_1; the volume of water expelled is then $(y_1 - y_0) \times 4 \times$ pipe area, this requiring an interval of 2 sec. The corresponding pipe velocity is therefore $2(y_1 - y_0)$ which is the horizontal distance between B_3' and B_3''.

By a similar method the distances B_5'–B_5'', and so forth are determined and the diagram constructed. Of course, care must be taken to distinguish between H_B and H_D although for many problems they may be equated, which saves much inconvenience in the work. When the flow reverses it is necessary to have the curves for the pump both when operating as a brake (that is without backward rotation of the impeller) and also as a turbine (where the impeller runs backward), but these problems need not be considered here.

On the same diagram the pressures and velocities for the pipe without an air chamber are found from the dashed lines, and a comparison of the two results shows that in this case the air chamber is of little value and a larger air chamber would make little difference. The illustration has been given primarily to show how the graphical method may be used.

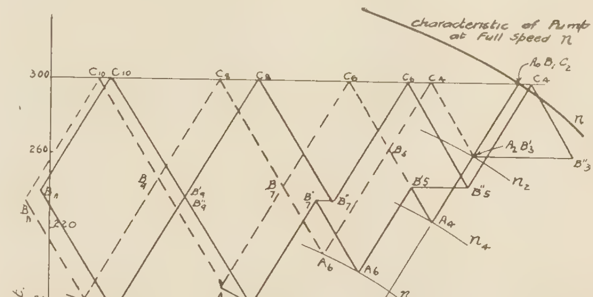

FIG. 6 WATER-HAMMER DIAGRAM RESULTING FROM POWER FAILURE TO A PUMP DELIVERING INTO A LINE IN THE CENTER OF WHICH IS AN AIR CHAMBER

PUMP-DISCHARGE LINE WHERE THE WATER COLUMN BREAKS

This problem is illustrated in Figs. 7, 8, and 9 and represents a case, not unusual in practice, where the line is laid on a slope from A to B and then horizontal from B to C. The pipe has been assumed of uniform diameter and thickness so that a is the same from end to end and was taken as 3300 fps, the velocity in the pipe under steady conditions being 4 fps. These condi-

tions give a value $\rho = (3300 \times 4)/(64.32 \times 260) = 0.79$. Suppose the pump was working at steady delivery and the driving power suddenly failed, then the pump would begin to slow up at once and the pressures would fall at A and B and all other points along the line. At B the normal pressure in the pipe was 90 ft, and as long as the pressure at this point did not fall more

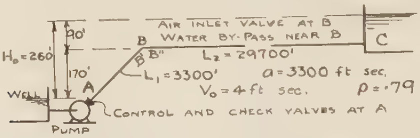

FIG. 7 PUMPING INSTALLATION WITH DELIVERY LINE PARTLY SLOPED AND PARTLY HORIZONTAL

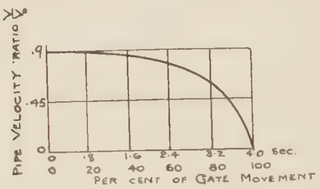

FIG. 8 ASSUMED DISCHARGE CHARACTERISTICS FOR A VALVE AT A IN FIG. 7

(When full open this valve permits a pipe velocity of 0.9 V_0 with a head of 260 ft on the pipe near A.)

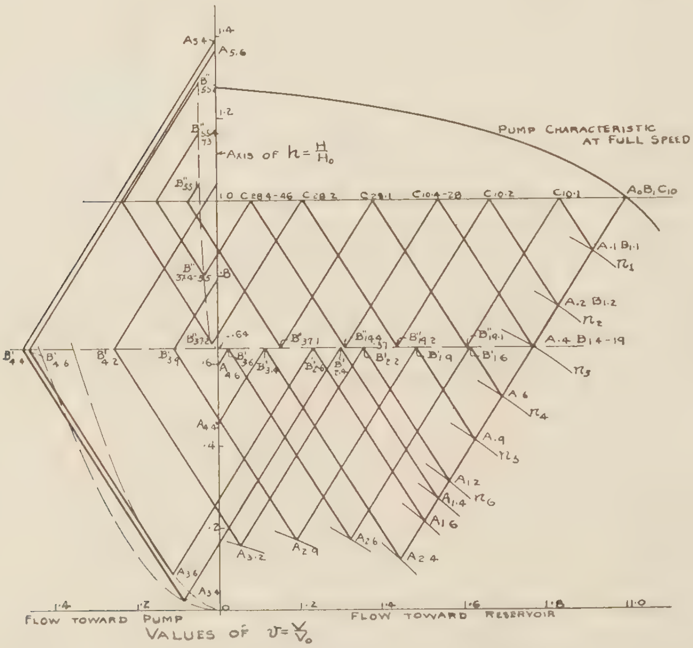

FIG. 9 WATER-HAMMER DIAGRAM RESULTING FROM SUDDEN FAILURE OF POWER SUPPLY TO THE SYSTEM SHOWN IN FIG. 7

than 90 ft the pressure at B would be positive. However, for a greater pressure drop the pressures would be negative reaching the maximum negative value of 34 ft for a drop of 124 ft; for this drop in pressure the water column would part for a time, but later on the columns would reunite and cause very serious, if not dangerous, water hammer.

It would be good practice to prevent any vacuum exceeding say 4 ft at B, and if an air-inlet valve were put there this condition could be maintained. That is to say, when the pressure at B fell 94 ft, or when h_B reached the value $h_B = (260 - 94)/260 = 0.64$, air would be admitted at B and the column would part there, the constant pressure at this point being 4 ft below atmosphere. The water-hammer diagram, using axes of h and v, has been drawn on Fig. 9 along with the characteristic curve of the pump at full speed, so that $A_0B_1C_{10}$ is located, and the line $h = 0.64$ is drawn as shown. From a knowledge of the pump's characteristics it is not difficult to compute the speed at any number of seconds after power failure, and thus a set of head-discharge curves may be drawn by well-known methods; the one at speed n_3, reached in 0.4 sec, which crosses the water-hammer line at $h = 0.64$ has been found by trial. A number of these pump curves have been drawn on the diagram, with the speed corresponding to each, and the number of seconds required for the pump to fall to this speed is marked on each curve. This locates A_0, $A_{0.1}$, $A_{0.2}$, and $A_{0.4}$, and so on, also B_1, $B_{1\cdot1}$, $B_{1\cdot2}$, and $B_{1\cdot4}$, and since the pressure at B is fixed at $h = 0.64$, the point $B_{1\cdot4}$ is also B_{19}, since the latter point lies on the line through C_{10}. The separation occurs in 0.4 sec.

The two columns now pursue independent courses, and referring first to BC, the water-hammer lines are drawn above $h = 0.64$ as indicated and the free end toward B of this column retreats a distance $(0.76 + 0.30) \times V_0 \times 18$ sec $= 76.3$ ft from the bend. On the other hand, the end toward B of the column AB passes the bend by $(0.76 + 0.09) \times V_0 \times 2$ sec $= 6.8$ ft, then returns, and a space containing only air exists in the pipe; its length is 76.3 ft. The direction of flow in BC would then reverse and the water would gain considerable speed before filling the space again, but when it did catch up to the column in AB the shock would be very severe. If a reservoir containing enough water to fill the space were available at B, then on the return the water columns would not acquire any serious velocity and consequently water hammer could not produce a pressure at B in excess of $\{1 + (1.00 - 0.64)\}H_0 = 1.36\ H_0$, which would not be dangerous. The structure necessary to provide this water would be rather expensive and difficult to execute, partly because of the large volume of water to be admitted quickly, and in general the plan is not justified.

Any attempt to use the air between the columns as a cushion is also ineffective, as will readily be seen on examination.

However, the use of properly designed valves at A and B, at relatively low cost and small inconvenience, would eliminate all of the danger.

Dealing first with the section BC and the part near B, suppose that a valve was placed between B and C but so close to B (somewhat more than 76.3 ft from it) that water always flowed through it, i.e., it would be as close to the final end of the column as possible. Of course it should be placed on a part of the line rising toward B. Fig. 9 shows that water would continue to flow toward the reservoir for 37.1 sec, after which it would begin to flow toward the pump. The placing of a valve which would close so as to prevent any return flow of water toward the pump, would prevent a pressure rise at the reservoir side of the valve in excess of $(1.00 - 0.64)\ H_0 = 0.36\ H_0$. The operation of this valve becomes quite simple in a long line such as is described here, because closure would actually be slow, and even if a small volume of water escaped past the valve the pressure rise could be kept very low; if difficulty were found in providing a suitable valve, one could be used which closed before the flow toward the reservoir had completely stopped, provided a much smaller by-pass valve was installed to let water through toward C; the small valve would then close quickly to prevent return flow.

The dashed parabola above $h = 0.64$ in Fig. 9 corresponds to a by-pass without a valve and with a discharge at 94-ft head equal to 4 per cent of the pipe. Such an arrangement gives the pressures $B''_{37.4-55}$, $B''_{55.1}$, and so on, the highest being $B''_{55.2}$ corresponding to $h = 1.28$. Return flow at any considerable velocity should be avoided.

For the lower column AB a valve would be necessary at A and if possible it should close at the instant reverse flow began, in which case the pressure rise at A would not exceed $0.64\ H_0$ above 166 ft, i.e., the total pressure at A would not exceed $166 + (0.64 \times 260) = 332$ ft. However, the section AB is relatively short, having a value $L/a = 1$ sec. Consequently, the closure of the valve at A would have to be quick and it would need to be very accurately timed or else high return velocity, accompanied by severe shock, would be attained. Therefore, it will be necessary to give this further study.

In the first place the discharge characteristics of a proposed valve would have to be known, and for lack of better data a curve for a valve has been assumed in Fig. 8; this is quite similar to that for a common gate valve, and the diagram is drawn on the axes of relative pipe velocity against distance traveled (or angle turned) by the valve, the head on the valve remaining constant at $H_0 = 260$ ft. It will now be assumed that the valve is closed at uniform rate in 4 sec after power failure so that the base represents both time and distance or angle traveled by the valve. From Fig. 8 a series of parabolas was drawn on corresponding to the times for the values of A noted on Fig. 9 and the water-hammer lines have been drawn there. These show that with the law of closure adopted here the pressure rise at A would not exceed 40 per cent which would occur 5.4 sec after power failure and could not be considered dangerous ordinarily. It is assumed that this valve is above a quick-closing check valve, and discharges into the pump.

In a case like this the reverse procedure might be adopted, that is, the maximum water pressure to be allowed might be decided upon and then a valve-closure law could be drawn to fit the case. Almost any form of valve-closure law may be obtained; for example, the valve may be closed by servomotor and the oil supply to this may be so regulated by pilot valve and cam as to fit any case desired in the same manner as used in water-turbine governors; however, a much simpler construction can be used here.

Water Hammer Due to Chattering or Slamming of Valves

This problem should be considered in the present series as it affects pumping lines and all piping, often seriously. A very large number of valves of various makes have a loose connection between the stem or operating spindle and the piece that performs the function of changing the area; for example, the disks of gate and globe valves are always free on the stems of the valves, for obvious reasons. Other valves having a rigid connection have a slight play of spindle or stem in its bearings or spindles which bend under pressure.

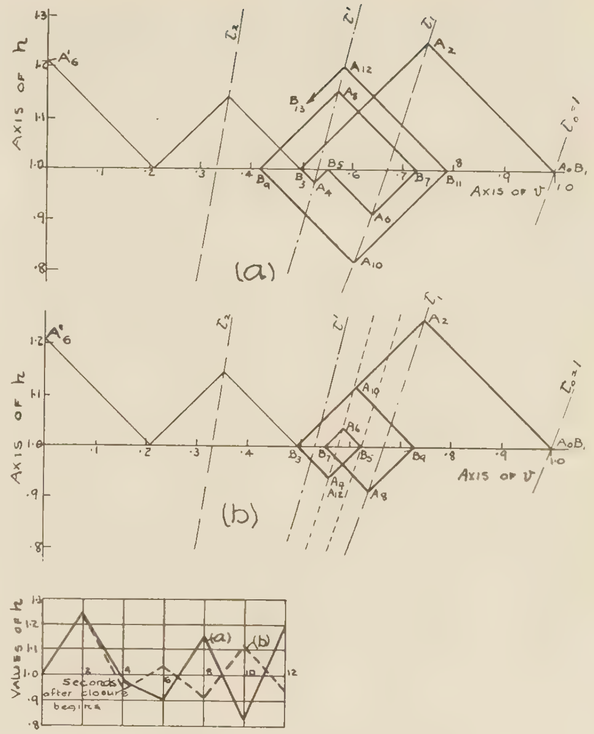

Fig. 10 Water Hammer Due to the Chattering of a Valve When One Third Closed

[(a) When the period of oscillation of the valve is the same as that of the pipe; (b) when the period of the valve is different from that of the line.]

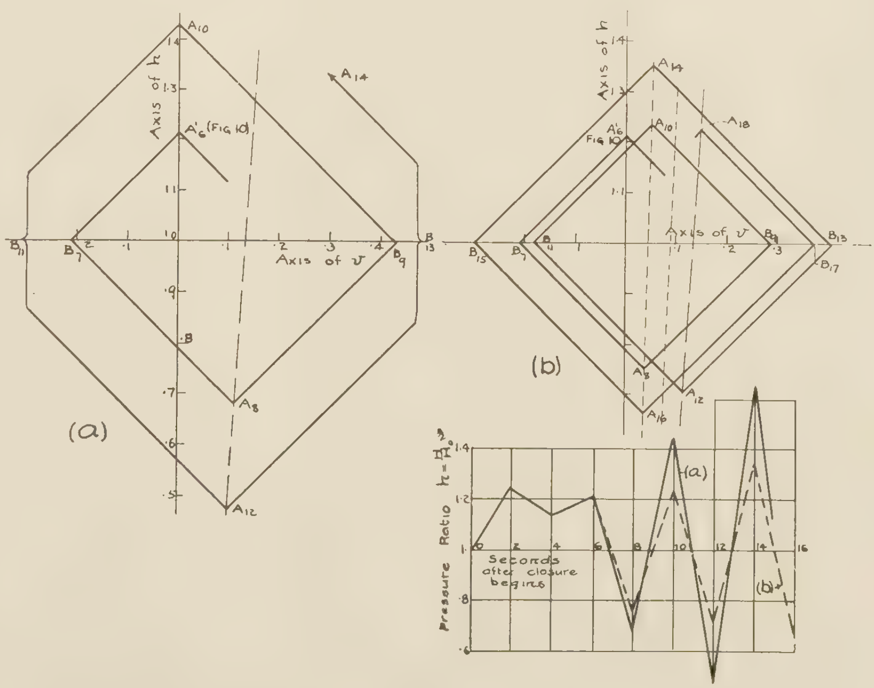

Fig. 11 Water Hammer Due to Chattering of a Valve When Nearly Closed

[(a) When the period of oscillation of the valve is the same as that of the pipe; (b) when the period of the valve is different from that of the line.]

All such valves are subject to chattering for certain settings, sometimes when partially closed and quite frequently when nearly closed; in the latter case, very rapid repetitions of closure and slight opening may occur or else the valve may slam down on its seat. Whichever happens, rapid, though possibly slight, velocity changes occur. A study of their effect will now be made, and for simplicity the case of uniform value of a and constant-level reservoir will be assumed. The first case will be one of chattering of a partly closed valve as shown in Fig. 10, where complete closure would occur in three intervals but where the valve has been stopped at the end of the first interval and chattering of the valve allows the discharge to fluctuate between the values shown by the parabolas τ_1 and τ'; the distance between these is abnormally large in order to clarify the drawing.

Case (*a*). Where the period of oscillation of the valve is the same as that of the pipe, resonance occurs with the result shown in Fig. 10(*a*). The exact shape of this diagram depends, of course, on the value of ρ, and on the setting of the valve at which the chattering occurs.

Case (*b*). Where the period of the valve is different from that of the line, as is not unusual, the diagram will be as shown at Fig. 10(*b*). In this case the valve has a period one half longer than the pipe, and the corresponding diagrams have been plotted on this assumption. It is for the same general data as Fig. 10(*a*) drawn to the same scale.

The next diagrams, shown in Fig. 11, are similar to those of Fig. 10, but are for cases where chattering occurs when the valve is nearly closed. The diagrams in Figs. 11(*a*) and 11(*b*) are for the valve periods shown on Figs. 10(*a*) and 10(*b*). "Slamming" of the valve under similar circumstances cannot produce greater water pressures than chattering, although of course the mechanical shock on the seat may not be the same.

In all of these cases the pressure builds up rapidly.

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Hydraulic Phenomena in Fuel-Injection Systems for Diesel Engines

By KALMAN J. DeJUHASZ,[1] STATE COLLEGE, PA.

The author compares the problems of water-hammer phenomena with those of fuel-injection surges. He also presents a graphical method for analyzing the pressure and velocity conditions and shows how it is applied to representative examples of injection systems with a timed valve, and those with a timed pump. In the latter group the action of "open" and "closed" nozzles is explained. Singular cases of injection conditions are analyzed. The effect of enclosed volume and of included rigid masses is examined. Various methods of terminating the injection are discussed.

FUEL OIL is introduced into the cylinder of the solid-injection Diesel engine in accurately measured doses in the form of a fine spray. This involves a frequent starting and stopping of the liquid flow which gives rise to transient phenomena closely akin to the water hammer occurring in hydraulic conduits. These phenomena greatly influence the characteristics of the injection and thereby influence the performance of the engine. Therefore, a knowledge of them is important for the rational design of fuel-injection systems.

For satisfactory engine performance the injection system must accomplish accurate metering and correct timing, and must consistently discharge at a constant rate. The system should also permit variability in timing and metering as the load and speed of the engine vary.

Comparison of Injection Systems With Hydraulic Conduits

In so far as injection systems and hydraulic conduits involve a disturbance in the equilibrium condition of a liquid flow, the injection process and the surges in hydraulic conduits are related phenomena obeying the same basic natural laws. However in hydraulic conduits attention is focused on the forces and stresses to which the structure is subjected, while the velocity and quantity of discharge during the disturbance of flow are mostly of secondary importance. In fuel-injection problems the opposite is the case. The time rate of discharge, the velocity and pressure changes, and their consistency or inconsistency between successive injections are the features of primary importance, while strength considerations are of secondary importance.

The range of pressures, velocities, time elements, and pipe dimensions in the two groups of phenomena are wide apart. In some respects the conditions in the injection systems are simpler. Thus, the wall thickness of the pipes employed is so large in comparison to the diameter that the pipes can be considered rigid, the speed of propagation of a disturbance depending only on the liquid and remaining constant all along the pipe. On the other hand, into an injection system several elements, such as storage spaces and moving rigid masses (spring-loaded nozzle valve), are embodied which exert an influence on the resulting flow phenomena. For all these considerations the equations, diagrams, and methods developed for water hammer are not suited for fuel-injection problems in their original forms, but have to be adapted to the particular conditions and purposes characterizing the fuel-injection systems.

Purpose and Method of Analysis

After the type of injection system has been chosen from considerations of the service to which the engine is to be placed and the main dimensions (plunger diameter and lift, length and diameter of the pipe, type and size of the nozzle, injection pressure to be used, and a number of other design data) have been determined by calculation or assumed on the basis of past experience, the designer can then determine (1) the metering, (2) the timing, and (3) the rate of discharge for those engine-operating conditions, i.e., (*a*) load and (*b*) speed, at which the engine will be mostly used and for which the optimum performance is desired. This done, the designer wants to know: (*A*) How the factors (1), (2), and (3) are influenced when factors (*a*) and (*b*) are changed; (*B*) what the effect on (1), (2), and (3) will be if the pipe length, pipe diameter, nozzle area, and injection pressure are altered in comparison with the initially assumed or determined values; and (*C*) how the consistency between successive injections will be influenced. These questions can be answered by means of an analytical or graphical analysis.

In this paper a graphical method of analysis is used, based on the fundamental considerations and natural laws as are used in dealing with hammer phenomena. Friction in the pipe due to viscosity and the effects due to leakage are neglected. Briefly, the method consists in the determination of the flow coordinates (velocity in the pipe and pressure in the pipe) for successive time-intervals of reflections in a v-p diagram, and in the determination of the time and location in the pipe at which a given flow condition exists in a t-x diagram. From these diagrams, two solid diagrams (stereograms) can be constructed, one for the velocity v and the other for the pressure p, both erected over the time-distance (t, x) plane as a base. With the aid of this representation a mental image can be formed of the seemingly complicated changes of the four coordinates v, p, t, and x; and a clear visualization of the influence of alterations on the originally assumed conditions is possible.

A full explanation of this method has been given in a previous

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Contributed by the Hydraulic Division for presentation at the Second Water-Hammer Symposium, in cooperation with the American Society of Civil Engineers and the American Water Works Association, at the Annual Meeting of The American Society of Mechanical Engineers, to be held in New York, N. Y., December 6–10, 1937.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until March 10, 1938, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

paper[2] by the author and therefore it will be omitted in this paper.

A number of representative examples will now be treated in some detail.

Injection Systems With Timed Valves

An injection system with a timed valve consists essentially of a storer of volume R, a nozzle of area f, and a connecting pipe of area F and length L. The nozzle is opened and closed by a cam-actuated valve driven from the engine shaft. The injected quantity is regulated either by (1) keeping the storer pressure substantially constant and altering the duration of injection, or (2) keeping the duration of opening constant and altering the storer pressure. Hydraulically, both systems bear a basic similarity to one another.

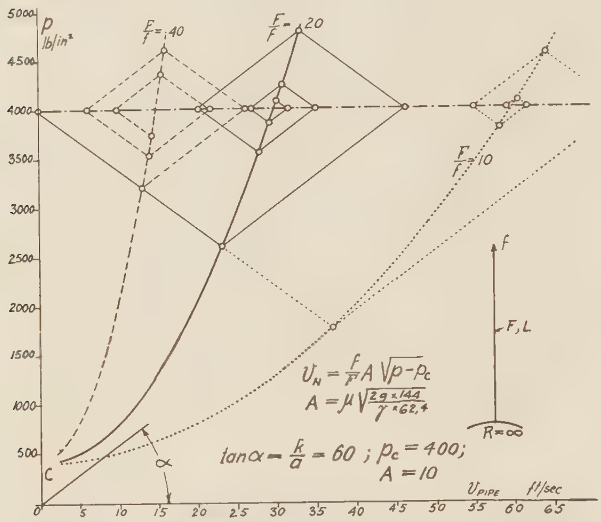


Fig. 1 Chart of Timed-Valve Injection With Infinite Storer for Various Ratios of Pipe Area to Nozzle Area

The graphical analysis of such a system is given in Fig. 1. A storer pressure of 4000 lb per sq in. and a pressure in the engine cylinder of $p_c = 400$ lb per sq in. are assumed. A fuel oil with a modulus of elasticity $k = 276{,}000$ lb per sq in. and an acoustic velocity of $a = 4600$ fps is assumed, giving a k/a ratio of 60. Constructing a v-p chart the directrix with $\tan \alpha = 60$ is drawn. The storer is assumed to have a large volume ($R = \infty$), so the pressure drop due to the efflux of liquid is neglected. Assuming an F/f ratio of 20, the nozzle characteristic (velocity in the pipe at the nozzle end) can be constructed from the efflux equation

$$v_N = \frac{f}{F}\,\mu \sqrt{\frac{2g \times 144}{\gamma \times 62.4}}\,\sqrt{p - p_c} = \frac{f}{F}\,A\,\sqrt{p - p_c} \ldots [1]$$

where

$$A = \mu \sqrt{\frac{2g \times 144}{\gamma \times 62.4}}\ (=)\ 10 \ldots\ldots\ldots\ldots [2]$$

This parabola is drawn in the v-p chart as a solid line. For the sake of simplicity an instantaneous opening and closing of the nozzle is assumed. Furthermore, the variation of the back pressure in the engine cylinder during the injection period is neglected. Actually, owing to the progress of compression and combustion, the back pressure undergoes a change with reference to its assumed constant value of $p_c = 400$; but, in view of the assumed high injection pressure such variations are of secondary importance.

On the basis of these simplifying assumptions, the efflux velocities and pressures are defined by points on the efflux parabola, obtained by intersecting it with lines of $\pm \tan \alpha$ as shown on the chart. The stereograms of velocity and of pressure are shown in axonometric representation on Fig. 2, from which the conditions of flow for all points in the pipe and for all instances of time can be clearly visualized. For the purposes of the study of injection, however, the history of velocity v_N and of pressure p_N at the nozzle end of the pipe is sufficient, and this is shown in Fig. 3, each value of v_N and p_N being valid for a time element of $2L/a$. The line v_N also represents, on a different scale, the rate of discharge.

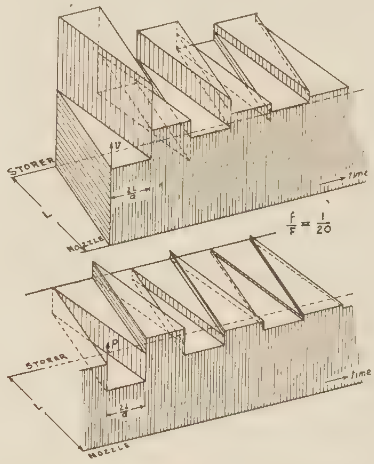


Fig. 2 Stereogram of Velocity and of Pressure for Timed-Valve Injection, According to Fig. 1 for $F/f = 20$

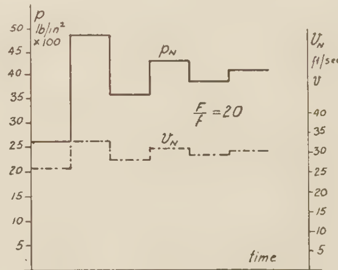


Fig. 3 History Diagram of Pressure and Velocity at the Nozzle End of the Pipe

The conclusions to be drawn from these diagrams are as follows: The efflux velocity converges toward the value of stationary flow corresponding to the storer pressure, the velocity of the successive surges being alternately above and below this limiting velocity. Shortening the pipe shortens the duration of the L/a interval, and therefore increases the number of surges in a given time. This brings the mean velocity of injection closer to the stationary value corresponding to the storer pressure.

Increasing the pipe area F and keeping the other variables constant can be represented in the chart by constructing efflux parabolas corresponding to the altered F/f ratio. This has been done in Fig. 1 for the F/f ratios of 10 and 40. It is seen that with the wider pipe the pressure fluctuations are of less absolute magnitude but converge slower toward the static value than is the case with a narrower pipe.

[2] "Transient Phenomena in Linear Flow," by K. J. DeJuhasz, *Journal of The Franklin Institute*, vol. 223, April, 1937, pp. 463–493; May, 1937, pp. 643–654; and June, 1937, pp. 751–778.

Instead assuming instantaneous opening and closure, the flow phenomena could be depicted in the v-p diagram also for gradual opening and closure corresponding to the actual manner in which such operations are performed. For each time-element in this case, the efflux parabola corresponding to the nozzle area ratio F/f, valid for that respective time-element, has to be constructed. The efflux points will then lie not on one parabola, but instead each one will lie on a different parabola valid for its respective time interval.

Futhermore, in the v-p diagram the changing of the back pressure could also be given consideration. In this case, the points would again lie on different parabolas, all of the same shape, but with their apexes located on that p_c point which corresponds to the back pressure existing in the cylinder at that particular time interval.

Another important characteristic of such an injection system is its consistency. After the injection valve is closed (which is effected by the cam in a positive manner) the injection definitely ends. The surges continue in the pipe but can be assumed to be damped out before the next injection begins. The next injection starts from the same initial conditions, provided the storer pressure is kept constant.

In actual cases, the storer does not have an infinite volume; therefore, the storer pressure is reduced by the injection. The reduction of pressure during one L/a interval is proportional to the velocity of efflux. The storer characteristic is not a $p =$

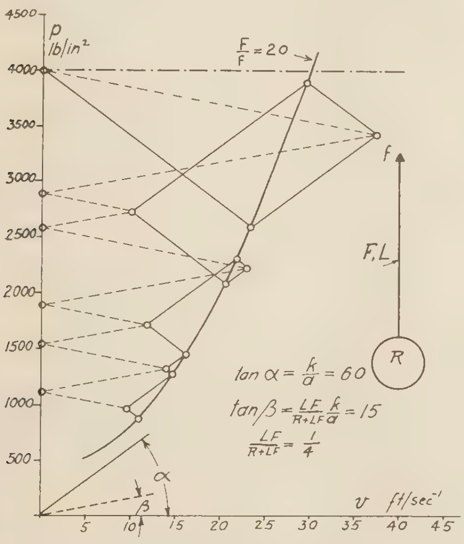


Fig. 4 Chart of Timed-Valve Injection With Finite Storer

onst line as in the case previously dealt with, but as pointed ut by the author in a previous paper[2] has a slope of

$$\tan\beta = \frac{LF}{R+LF}\,\frac{k}{a} \qquad [3]$$

he construction of the v-p chart for such a case is shown in Fig. 4, assuming a ratio

$$\frac{LF}{R+LF} = \frac{1}{4}$$

Different Types of Pressure Variations

Referring again to the case of an infinite storer, Fig. 1, it is een that the pressure at the nozzle in alternate $2L/a$ intervals is rst below and then above the storer pressure. This is not necessarily the case, however, since several types of pressure variations are possible according to the magnitude of the storer pressure, as shown in Fig. 5.

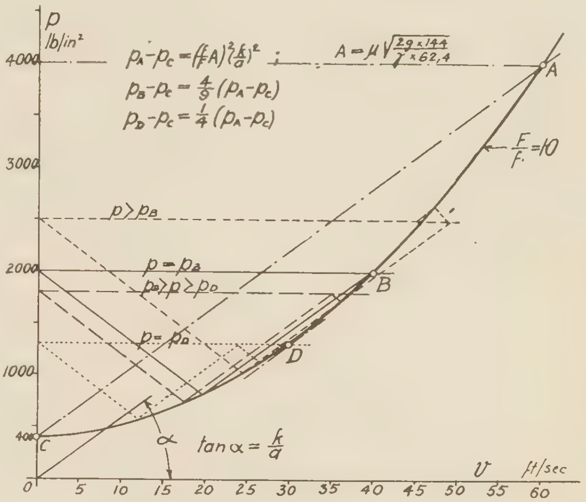


Fig. 5 Chart of Timed-Valve Injection for Particular Values of Storer Pressure

A critical point on the efflux parabola is A, the intersection with the k/a directrix drawn from the apex, the coordinates of which can be expressed as

$$v_A = \left(\frac{f}{F}A\right)^2 \frac{k}{a} \qquad [4]$$

and

$$p_A - p_c = \left(\frac{f}{F}A\right)^2 \left(\frac{k}{a}\right)^2 \qquad [5]$$

In the present example $v_A = 60$; $p_A - p_c = 3600$; and $p_A = 4000$.

Case I. There is a critical pressure p_B, corresponding to point B on the efflux parabola, at which the second surge restores the pressure to the full value of the storer pressure, and efflux takes place henceforth with the stationary velocity corresponding to the storer pressure. The coordinates of this flow condition are

$$v_B = \frac{2}{3}v_A \qquad [6]$$

and

$$p_B - p_c = \frac{4}{9}(p_A - p_c) \qquad [7]$$

In this example $v_B = 40$; $p_B - p_c = 1600$; $p_B = 2000$.

Case II. The coordinates of the point D, at which the tangent of the efflux parabola is a directrix, can be expressed as

$$v_D = \frac{1}{2}v_A \qquad [8]$$

and

$$p_D - p_c = \frac{1}{4}(p_A - p_c) \qquad [9]$$

In this example $v_D = 30$; $p_D - p_c = 900$; $p_D = 1300$.

The following cases can be distinguished:

Case III. If the storer pressure $p \leqq p_D$, then the efflux pressure never exceeds the storer pressure.

Case IV. If the storer pressure $p_D < p < p_B$, then in the later intervals of efflux the efflux pressure exceeds the storer pressure.

Case V. If the storer pressure $p > p_B$, then in every alternate double interval of efflux, the efflux pressure exceeds the storer pressure.

These cases can be applied also to hydraulic reservoirs, penstocks, and gates.

Injection Systems With Timed Pump

Essentially such systems consist of a pump characterized by its rate of delivery, of a nozzle of area f, and a connecting pipe of length L and area F.

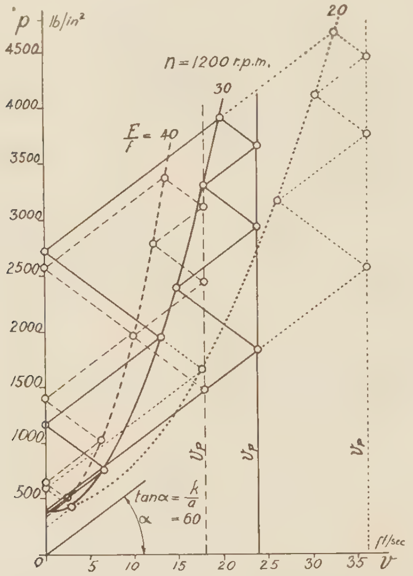


Fig. 6 Chart of Pump Injection for Various Ratios of Pipe Area to Nozzle Area (F/f = 40, 30, and 20), for an Open Nozzle

Fig. 6 represents the graphical analysis of such a system. A cylinder back pressure of 400 lb per sq in., a fuel oil with a ratio of k/a = 60, and a ratio of F/f = 30 are assumed.

For an engine cylinder of 4.375 in. bore and 5.25 in. stroke (a volume of 79 cu in.), the fuel to be injected at full load is 0.004 cu in. Assuming an engine speed of 1200 rpm and a duration of pump delivery of 20 deg crank angle, the time duration will be 0.002775 sec. With a pipe length L = 2.12 ft, the interval of single traverse will be L/a = 0.00046 sec, so that the period of pump delivery is equivalent to $6(L/a)$. Other data are: Plunger diameter = 0.275 in.; plunger area = 0.0594 sq in.; effective plunger travel = 0.066 in.; pipe diameter = 0.08 in.; pipe area = 0.00503 sq in.; nozzle diameter = 0.0145 in.; nozzle area = 0.0000165 sq in.; and the ratio of plunger area: pipe area:nozzle area = 360:30:1.

The performance of this system under the foregoing specified conditions, together with deviations from these conditions, will be investigated.

Open Nozzle

In Fig. 6 the v-p diagram is given for n = 1200 rpm and for pipe areas corresponding to F/f = 40, 30, and 20. The pressures and velocities at the nozzle are shown as a function of time in Fig. 7.

For a complete visualization of the flow, two solid diagrams, one for the velocity and one for the pressure, have been constructed and are shown in Fig. 8. Comparing the effect of altering the pipe area, it is seen that the narrowest pipe gives the most rapid pressure rise during delivery and the most rapid decrease of pressure after delivery has ceased.

Fig. 9 shows the effect of changing the pump speed. Three speeds of 900, 1200, and 1800 rpm are considered. The history diagram of the velocity and of the pressure is given in Fig. 10 as a function of time and in Fig. 11 as a function of crank angle. It is seen that, in terms of time, the velocity of discharge increases with increasing speed. In terms of crank angle, however, the injection of a given amount of oil takes a larger crank angle at high speeds. The chart shows also the increase of the injection

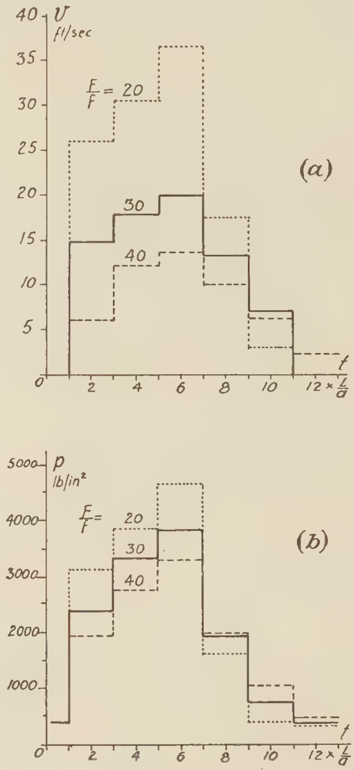


Fig. 7 History Diagrams of (a) Velocity Versus Time and (b) Pressure Versus Time for Pressure and Velocity at the Nozzle End of a Pipe

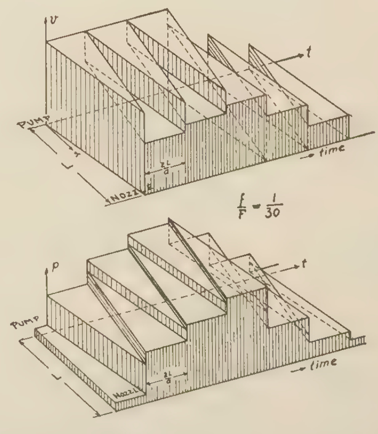


Fig. 8 Stereogram of Velocity and Pressure for Timed-Pump Injection, According to Fig. 6 for F/f = 30

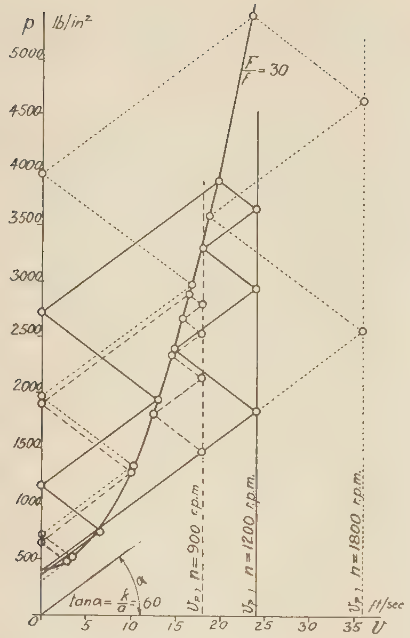


FIG. 9 CHART OF PUMP INJECTION FOR OPEN NOZZLE AND FOR VARIOUS PUMP SPEEDS. SPEEDS OF 900, 1200, AND 1800 RPM ARE INDICATED

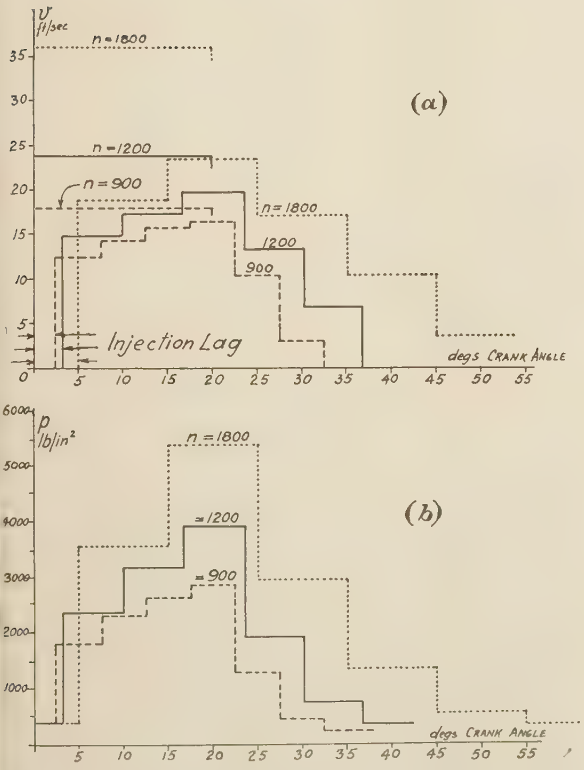


FIG. 11 HISTORY DIAGRAMS OF PRESSURE AND VELOCITY ON CRANK-ANGLE BASIS AT THE NOZZLE END OF A PIPE

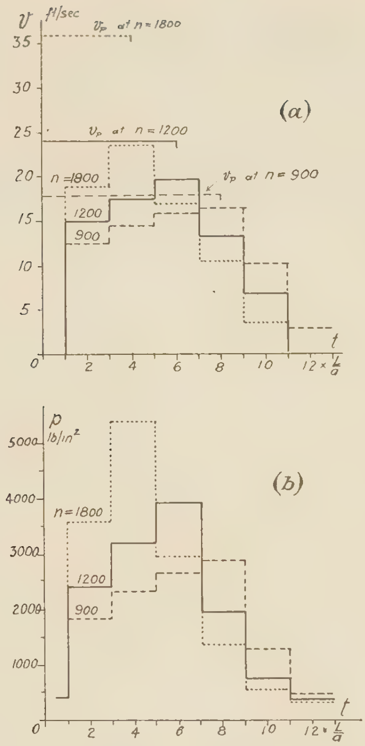


FIG. 10 HISTORY DIAGRAMS OF PRESSURE AND VELOCITY ON TIME BASIS AT THE NOZZLE END OF A PIPE

lag (elapsed crank angle between the beginning of pump delivery and the beginning of nozzle discharge) with increasing speed.

AFTER-DRIBBLING

Again, in this analysis, the change of back pressure during the period of injection is neglected. This does not affect much the

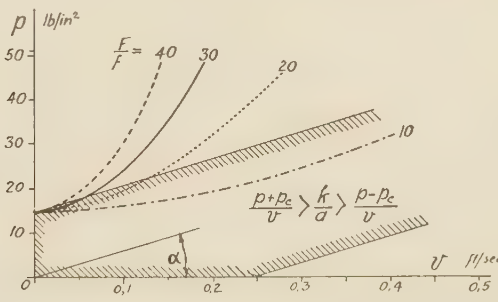


FIG. 12 CONDITIONS FOR RAPID CUTOFF OF INJECTION WITH TIMED PUMP AND OPEN NOZZLE

flow phenomena in the vicinity of the compression end of the stroke during which the bulk of the charge is injected, but it has an important influence during the end of the expansion stroke when the cylinder pressure is near atmospheric pressure. Fuel which issues from the nozzle at low pressure (and hence at low speed) is not atomized sufficiently, and it has a detrimental effect on the nozzle orifice (clogging, formation of carbon deposit). This phenomenon is called "after-dribbling." In seeking the criterion of sharp cutoff of injection, it must be borne in mind that no more outflow will take place after the liquid in the pipe has attained zero velocity. This criterion is illustrated in Fig. 12.

Whenever the velocity and pressure of the liquid column corresponds to a point in the shaded area in Fig. 12, and is reflected toward the plunger (which is standing still), there will be no more efflux from the nozzle. This consideration favors the use of narrow pipe (F/f small), as is evident from Fig. 12.

Different Types of Velocity Variations

According to the magnitude of the rate of pump delivery several types of velocity variations in the efflux are possible, as indicated in Fig. 13.

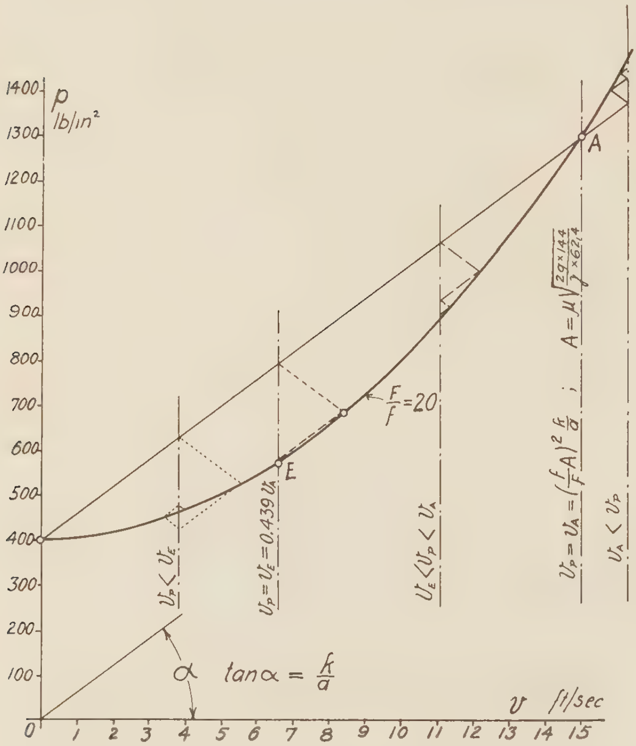


Fig. 13 Chart of Pump Injection for Particular Values of Plunger Velocities

Case I. If the rate of delivery is such that the velocity in the pipe at the pump is equal to the v_A value (the coordinates of the point A are given in Equations [4] and [5]), then there will be no reflection at the nozzle, and the efflux velocity will be the same during the total period of injection. In this example v_A = 15; $p_A - p_e = 900$; $p_A = 1300$.

Case II. Another critical point on the efflux parabola is E for which equilibrium condition is attained with the second surge, after which no further reflection takes place. The coordinates of this point are

$$v_E = 0.439\, v_A \qquad [10]$$

and

$$p_E - p_e = 0.193\,(p_A - p_e) \qquad [11]$$

In this example $v_E = 6.6$; $p_E - p_e = 174$; $p_E = 574$.

Case III. If the rate of pump delivery were such that $v_P \geqq v_A$, then the efflux velocity would never exceed the rate of pump delivery.

Case IV. If the rate of pump delivery were such that $v_A > v_P > v_E$, then the efflux velocity would exceed the rate of pump delivery in all intervals of pump delivery.

Case V. If the rate of pump delivery were such that $v_E > v_P$, then the efflux velocity, in successive $2L/a$ periods, would be alternately greater and smaller than the rate of pump delivery.

Closed Nozzles

From the point of view of engine operation an important drawback of the open nozzle is the slow flow at the end of the injection period, resulting from the low pressure at which the last portion of the injection takes place. The closed-type nozzle was introduced to avoid this fault, and its various modifications have found an ever-increasing application in engine construction.

In this type of nozzle a spring-loaded valve keeps the nozzle orifice closed until the oil pressure attains the valve-opening pressure (*VOP*). Increasing oil pressure lifts the valve farther and thus increases the effective opening until the nozzle valve comes to a limiting stop. Above this pressure the nozzle acts virtually as a constant orifice. Now if the pressure be decreased, the valve will descend gradually and at the valve-closing-pressure (*VCP*), it comes to its seat again. Because of the difference of the area acted upon by the fluid pressure when the valve is closed, as compared to the area when the valve is open, a greater pressure is needed to open the valve than to keep it open; hence, the valve-opening pressure is greater than the valve-closing pressure. The characteristic of such a nozzle in the v-p diagram is represented by a parabola, the apical portion of which is truncated by two lines. The one originating from the valve-opening pressure

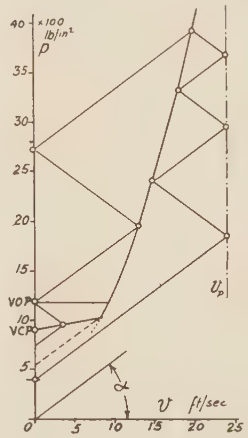


Fig. 14 Chart of Pump Injection With Closed Nozzle

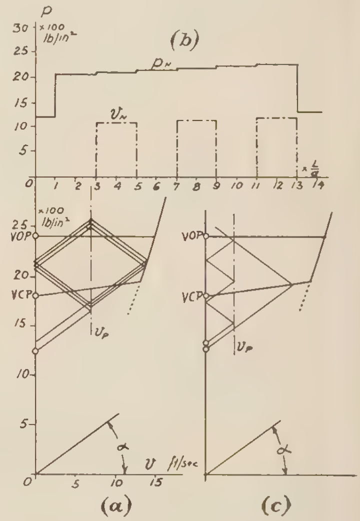


Fig. 15 Charts and Diagram of Pump Injection With Closed Nozzle

[(*a*) Chart of low plunger velocity with sustained delivery; (*b*) history diagram of pressure and velocity for (*a*); (*c*) chart of low plunger velocity with short delivery.]

must be used when the nozzle is not yet open, and the other, originating from the valve-closing pressure, must be used when the nozzle is already open. The graphical analysis of the injection lasting $6L/a$ intervals is given in Fig. 14. In this example a ratio of F/f = 30 is assumed in the full-open condition of the valve. The valve-opening pressure *VOP* = 1200, and the valve-closing pressure *VCP* = 900. It is seen that in comparison with the open nozzle, the slow dribbling end of the injection is cut away and the mean pressure of the injection is higher than in the case of the open nozzle. It is also seen that the pressure at the beginning of the injection (400 lb per sq in.) is not necessarily equal to the pressure at the end of the injection (750 lb per sq in.). Neglecting leakage between the periods of injections, the next injection is

defined by the directrix tangent to the injection characteristics, i.e., 580 lb per sq in. and the valve-closing pressure, i.e., 900 lb per sq in. Within this range, therefore, a variation is possible; and this defines the degree of consistency between the successive injections.

The phenomenon of buzzing or chattering of the needle valve may arise at a low velocity of pump delivery. This is illustrated in Fig. 15. For alternate $2L/a$ periods the nozzle is open and then again closed. If, in this case, one injection period lasts only for a $2L/a$ period of time or less, then every alternate injection will be missing, and the phenomenon of eight-stroking in the engine will take place as illustrated in Fig. 15*c*. With further lowering of the velocity of the pump delivery and decreasing its duration, the injection will be missed even more frequently. Thus, it is seen that in the case of a "closed" nozzle, there is a limit to the amount which can be injected regularly without missing. This amount depends on the velocity of pump delivery—at high speed the injectable minimum is less than at low speeds.

Effect of Enclosed Volume

In pump-injection systems, an enlarged volume adjoins the pipe which contains the spring for the delivery valve. Sometimes another enlarged volume is attached to the other end of the pipe (e.g., a fuel filter). These volumes act as storers and thereby modify the injection process.

Figs. 16*a* and 16*b* show the graphical analysis of an injection system containing a storer between the pump and the pipe. It is seen that the presence of the storer reduces the velocity of injection, and the approach to the pump rate of discharge is less rapid than in a system without such an enlarged volume.

Effect of Moving Rigid Masses

Rigid masses acted upon by the liquid pressure partake in the velocity change of the liquid column. The graphical analysis of

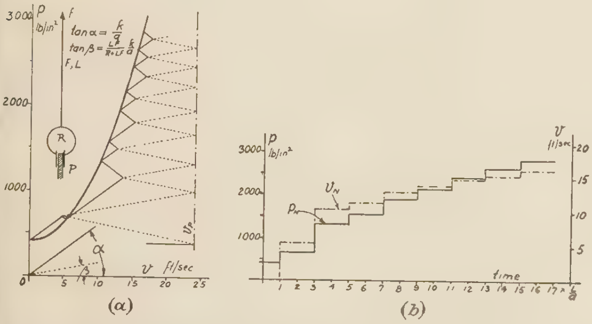


Fig. 16 Chart and History Diagram for Pump Injection With a Storer Adjoining the Pump

such elements, called "swingers," has been fully explained in another paper[2] by the author. The nozzle valve contained in the closed nozzle is such a swinger. In the present investigation only the time required for the fuel pressure to open the nozzle valve and whether the opening time is sufficiently long to importantly influence the injection process are considered. The following data will be assumed:

Nozzle-valve diameter $d = 0.25$ in., and area $f = 0.05$ sq in.; weight of nozzle valve and adjoining parts (stem, spring plate, and one third the weight of the spring) = 0.1 lb; mass of nozzle valve and adjoining parts, $m = 0.0031$; force acting on the nozzle valve at the moment it leaves its seat, $P = f(VOP - VCP) = 0.05\ (1200 - 900) = 15$ lb; acceleration $a = P/m = 4850$ fps per sec = 58,000 in. per sec per sec; travel of nozzle valve (limited by stop) $s = 0.02$ (assumed); and time needed for nozzle-valve

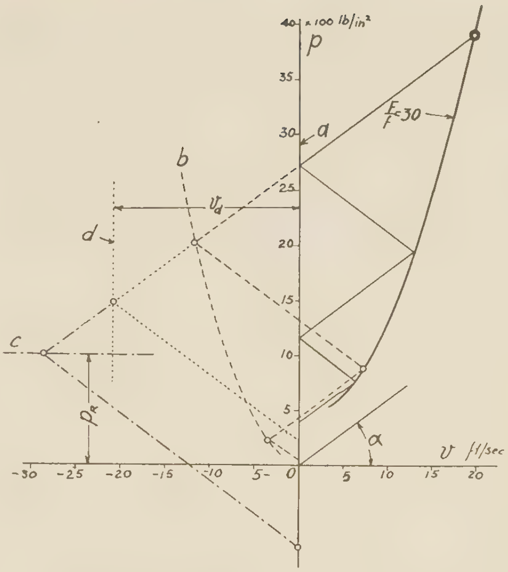


Fig. 17 Effect of Different Methods for Terminating the Injection as Represented on the v-p Chart

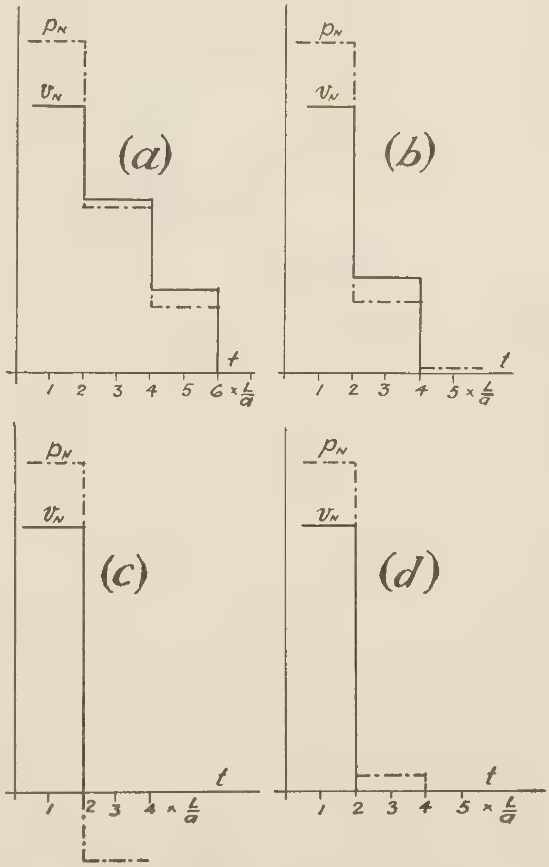


Fig. 18 Effect of Different Methods for Terminating the Injection as Represented on the History Diagram

[(*a*) The plunger comes to a standstill, or a delivery valve of negligible lift seats; (*b*) a small spill orifice is opened, no delivery valve is used; (*c*) a large orifice is opened between the pump space and a spill space under pressure, no delivery valve is used; (*d*) a retracting plunger (e.g., Bosch-type retracting valve) withdraws liquid at a constant rate.]

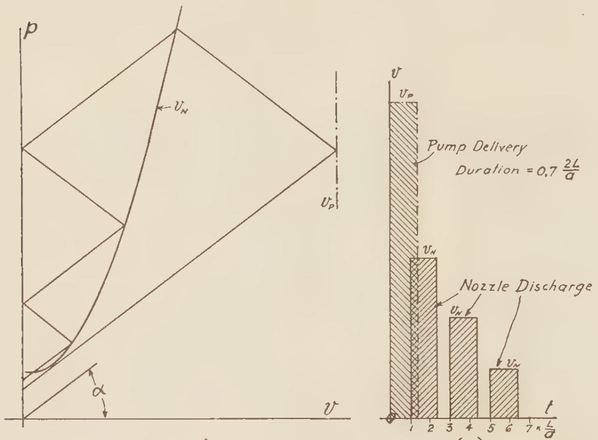
travel (assuming uniform acceleration) $t = 2s/a = 0.000835$ sec.

According to the conditions assumed in a previous example, the interval of single traverse was $L/a = 0.00046$ sec. Therefore, under similar conditions the full valve opening would be accomplished in less than 2 intervals, even at very low plunger velocities. At high plunger velocities, the pressure of the wave will greatly exceed the valve-opening pressure *VOP;* hence, the accelerating force will be correspondingly higher and the time of complete opening correspondingly shorter. Therefore, in the assumed example, the inertia of the nozzle valve will affect only the first and last two intervals of the injection.

Termination of Injection

The injection can be terminated in several ways, each of which can be suitably represented in the *v-p* diagram as shown in Fig. 17, and in the history diagrams shown in Fig. 18.

(*a*) *Termination by Stoppage of Flow at Pump.* This can be effected by a stoppage of the plunger, or by any other way of



(*a*) (*b*)

Fig. 19 Interrupted Discharge With Open Nozzle and Nonretracting Delivery Valve at a Pump Delivery Period Less Than $2L/a$. Two Interruptions Occur as Shown in (*b*)

terminating the pump delivery, provided a delivery valve of negligible lift is used. This method of termination is represented in Fig. 17 by the flow coordinates of the pump end of the pipe being located on the $v = 0$ line, that is, line *a*. In the previous examples this method of termination was assumed. The end of discharge is characterized by the pressure diminishing in steps, as shown by Fig. 18*a*.

(*b* and *c*) *Termination by Return Flow at the Pump.* This is effected by opening a spill-orifice which establishes a communication between the pump space and another space of lower pressure, provided no delivery valve is interposed between the pump and pipe. If the spill-orifice is small this method is equivalent to adding a second orifice at the pump end of the pipe, and the flow-coordinates will be located on the efflux parabola valid for this second orifice, as shown by line *b* in Fig. 17 and also by Fig. 18*b*. If the spill orifice is large, then it can be assumed that the pressure at the pump end of the pipe will immediately attain the p_R value of the spill space as shown by line *c* in Fig. 17, and also by Fig. 18*c;* therefore, the flow coordinates will be located on the p_R line. The injection comes to an end when the pressure at the nozzle end of the pipe sinks below the back pressure (in open nozzles), or below the valve closing pressure (in closed nozzles).

The final pipe pressure in both cases, *b* and *c*, is the pressure of the spill space. Therefore, both these cases would be operable only in conjunction with a closed nozzle, which acts virtually as the delivery valve and prevents the entrance of cylinder gases into the pipe. Because of the wide range of pressure fluctuations in the pipe and in the pump space, and to other engineering considerations, neither of these methods is used in practice, and both are of only academic interest.

(*d*) *Termination by Return Flow With Subsequent Stoppage of Flow at the Pump.* This can be effected by methods (*b*) and (*c*) with the interposition of a retracting type of delivery valve between the pump and the pipe. Actually every discharge valve with non-negligible lift has a certain retracting effect. This method is equivalent virtually to the plunger of the retracting valve withdrawing liquid from the pipe, the plunger motion depending on the pressure difference between the two sides of the valve. Assuming this velocity constant (actually it would be variable), its representation in the *v-p* chart is shown in Fig. 17 by line *d* and also in Fig. 18*d*. This method of termination is widely used in engineering practice.

In actual injection systems the method of terminating the injection is modified by the gradual rather than instantaneous opening of the spill orifice, by the pressure in the spill space being altered by the spilled liquid, by the varying velocity of the retracting valve, and by the other secondary influencing factors. Therefore, the foregoing four cases have been dealt with only in an approximate manner, and serve mainly to give a qualitative mental picture of the phenomena which occur.

Interrupted Discharge

Under certain conditions the termination of injection is characterized by a gap or break in the continuity of flow. Interrupted

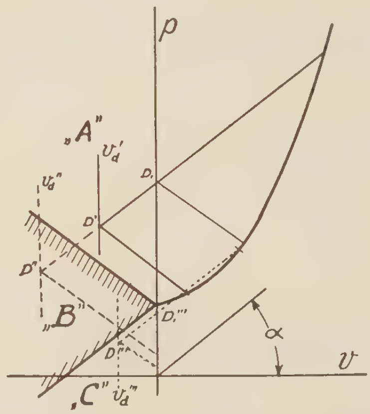


Fig. 20 Chart for Interrupted Discharge With Open Nozzle and Retracting Delivery Valve

[Interruption will occur if (1) the duration of the motion of the delivery valve is less that $2L/a$, and (2) first terminating wave D'' is located in area *B*.]

discharge is similar to the afterdribbling previously discussed. Both are detrimental to engine operation and should be avoided. With the aid of the graphical analysis, a complete explanation of these phenomena can be given. Only a brief outline of the analysis for the two methods of termination (*a*) and (*d*), discussed in the previous paragraph, will be given.

(*a*) *Termination by Stoppage of Flow.* Interruption can occur if the pump delivery lasts less than a $2L/a$ interval. If the velocity of pump delivery into the pipe is high enough, then several interruptions may take place. This case is represented in Figs. 19*a* and 19*b* for an open nozzle (two interruptions). For a closed nozzle similar considerations prevail.

If the pump delivery period is longer than $2L/a$ but not an inte-

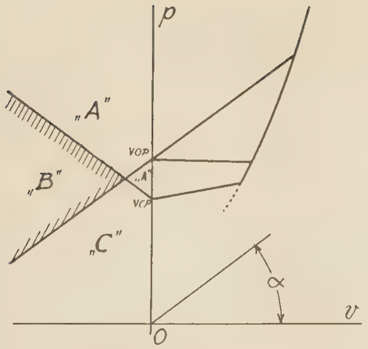


FIG. 21 CHART FOR INTERRUPTED DISCHARGE WITH CLOSED NOZZLE AND RETRACTING DELIVERY VALVE

ger multiple thereof, then the termination of the injection will be fluctuating, but there will be no interruption of flow.

(*d*) *Termination by Return Flow With Subsequent Stoppage of Flow.* Interruption of flow is possible even if the flow has been steady preceding the end of pump delivery. In the example, Fig. 20, an open nozzle and a "retracting" valve moving with a constant velocity for a length of time $\Delta t < 2L/a$ are assumed. Two waves of termination are thus initiated at the retracting valve, one characterized by point D (corresponding to the velocity of retracting) and another by point D_1 situated on the same directrix (corresponding to zero velocity). Depending on the location of the point D, the forward waves reaching the nozzle from the points D and D_1 may or may not produce flow, and accordingly three cases are possible:

1 If the point D lies in the territory A, then both waves initiated from points D and D_1 will produce flow from the nozzle, and there will be no interruption of discharge, that is, $D = D'$.

2 If the point D lies in the territory B then the wave initiated from D will not produce flow from the nozzle but the one from D_1 will, and thus there will be an interruption of discharge, that is, $D = D''$.

3 If the point D lies in the territory C neither the wave initiated from D''' nor the one from D_1''' will produce flow from the nozzle, and there will be no secondary discharge.

It may be noted that the territory B in Fig. 20 is bounded by the two directrixes intersecting the p-axis at the apex of the efflux parabola.

For the case of a closed nozzle, the territories A, B, and C, are shown on Fig. 21. It may be noted that the territory B is bounded by the two directrixes passing through the $v = 0$, VOP; and the $v = 0$, VCP points.

CONCLUSIONS

It has been shown that the graphical method presented in this paper is well-adapted for the analysis of problems of pressure and velocity waves in injection systems. The interplay of pressures and velocities occurring in these phenomena may be visualized, and the performance of a fuel-injection system may be predicted from its design. While simplifying assumptions have been made in the examples given, the method is capable of further refinement to approach more closely the conditions actually existing in practice. This method offers the possibility of reducing the amount of experimental work necessary for the development of a satisfactory fuel-injection system for a given service.

Operation of Emergency Shutoff Valves in Pipe Lines

By F. KNAPP,[1] SÃO PAULO, BRAZIL

At the principal control points of waterworks distribution systems, as well as in turbine penstocks, automatic valves are installed to shut off the flow of water in case of a pipe rupture. It is usual practice to assume the time and closure law of the valve and then to check the maximum surge caused by the valve closure.

It is the purpose of this paper to determine the characteristics and the hydraulic design of self-acting shutoff valves in such a manner that the water hammer, even in the worst emergency condition, remains within permissible limits. To reduce the discharge of water after a break, the law of closure is chosen in such a manner that this discharge becomes a minimum. The necessary computations are made by the graphical method of pressure-rise determination as well as by simultaneous or conjugated equations. The author assumes that the reader is familiar with these methods.[2,3,4]

Tripping Devices for Shutoff Valves

For operating the closing mechanism of a shutoff valve in case of a pipe rupture, a change of flow conditions is used to give the necessary impulse. These changes of flow conditions must reach values attained only in an emergency, but never reached in normal operation. The operating conditions of the pipe line are the deciding factor in the choice of the tripping device. The pressure and velocity conditions in a pipe line near the shutoff valve are influenced in varying degrees by a break in the line and must be examined in each specific case. As a rule, a considerable increase of velocity and a corresponding pressure drop will take place. Either a change of velocity or a change of pressure may be used to operate the tripping device, and the choice depends on whether, at the location of the automatic valve, a pipe rupture would cause first an appreciable change of velocity and then a change of pressure or vice versa.

Tripping devices depending on velocity changes consist of paddle trips, mercury contactors or a flow meter with maximum-flow contacts.

In other cases the rate of pressure change is greater than that of the velocity and pressure-operated switches and minimum-contact manometers are suitable; these are often used in combination with devices depending on velocity changes. In certain installations neither a velocity nor a pressure change is caused, for example, by a pipe rupture near the upper end of a pipe line leading from a pump house to a reservoir. In this case no appreciable changes of flow conditions at the instruments in the pump house are noticeable. For important installations flow meters are built in the pump discharge and at the reservoir. The automatic valve near the pump is closed if the difference between the discharge readings of the two meters exceeds a predetermined limit.

Venturi Meters Below Safety Valves

The case of turbine penstocks with a safety valve at the upper end of the line and a venturi meter immediately below deserves special attention. In case of a break below the venturi, the discharge through the meter with a great ratio of throat diameters may be limited to such an extent that the velocity-actuated tripping device does not operate. Such an arrangement upsets completely the purpose of the emergency shutoff valve.

Design Requirements for Safety Valves

The protection of main branches of waterworks distribution systems or of turbine penstocks is of considerable importance inasmuch as a rupture frequently causes extensive damage to property. Emergency closing valves designed without consideration to water-hammer considerations do more harm than good and may even be the cause of accidents. A correctly designed installation should fulfill the following conditions:

1 The maximum surge caused by closure of the valve in case of a rupture should not exceed a certain limit, determined by the strength of the pipes. This maximum pressure rise should not be influenced by quick closure or even slamming of the main valve.

2 The operation of the valve should be such that a minimum amount of water is discharged after rupture and while closing the valve.

These two design conditions are seldom met in the usual installation. It will be seen later that these requirements can be incorporated in emergency-valve installations with only slight modifications of the usual arrangement.

Type of Valves

Emergency closing valves should permit quick closing under unbalanced pressure conditions, and for this reason it is essential that the valve be free from vibrations under such operating conditions. The hydraulic forces should act in such a manner as to close the valve quickly. Reversal of these forces during the closing stroke is an undesirable feature.

Gate valves, besides having long strokes, require considerable power and require the longest closing time of all types of valves. This type of valve can only be installed where ample headroom is

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[2] "Water Hammer in Pump Discharge Lines," by O. Schnyder, *Schweizerische Bauzeitung*, vol. 94, nos. 22 and 23, 1929, pp. 271 and 283.

[3] "Variations of Flow in Water Conduits," by L. Bergeron, Comptes Rendu des Travaux de la Société-Hydrotechnique de France, 1932.

[4] "Simple Graphical Solution for Pressure Rise in Pipes and Pump Discharge Lines," by R. W. Angus, *English Journal* (Canada), 1935.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

available, and hydraulic operation is only possible where a separate source of pressure is available which is unaffected by the rupture of the pipe line.

Needle valves, because of their short stroke and nearly balanced pressure conditions, permit very short closing times and are usually free from vibrations. Unless supplied with a restoring mechanism, this type has a tendency to accelerate in the last few inches of travel to its seat.

Butterfly valves are essentially quick-closing devices and may be provided, like the needle valves, with electric or hydraulic power drive or may be operated by means of a heavy weight. The force required for closure is relatively small. Of all types concerned, butterfly valves require the smallest space.

The butterfly valve is the least expensive, the gate valve next, and the needle valve the most expensive.

Emergency valves may be subject to free-discharge conditions during closure and must, therefore, have a very rugged control mechanism of such a design as to permit easy adjustment of time and rate of closure. Cavitation in emergency valves is not a limiting feature, being subject to it only in the case of valve closure under free discharge which occurs at rare intervals.

Two Limiting Cases of Emergency Valve Operation

According to the location of the pipe rupture, the pressure and velocity conditions at the shutoff valve may be influenced to a variable degree. The distribution pipe lines, downstream from the valve, with its several branches and subbranches, are sometimes of such a complicated nature that even an approximate determination of pressure and velocity at the valve in case of an emergency condition becomes impossible. The actual flow conditions at the shutoff valve are always subject to two limiting conditions, i.e., (1) the pressure at the valve, in case of a break drops instantaneously to atmospheric pressure; and (2) the pressure remains constant.[5]

Any possible location of a pipe rupture and the water-hammer conditions caused thereby, are thus fully defined by these two limiting cases. The pressure variations, and consequently the velocity changes, will be greatest in the first case and, since the maximum surge will occur under these conditions, the operative characteristics of the shutoff valve are thus determined.

Proposed Design of Emergency Valves

The forces acting on automatic shutoff valves during closure are of considerable magnitude and, even with the most carefully designed control mechanism, the possibility of the slamming of the valve must be considered. It is of advantage to install, in addition to the quick-closing main valve, a pressure- or weight-operated by-pass valve which closes slowly after the main valve is closed. In order to limit the discharge after rupture, the main valve should close as quickly as possible. Closing times equal to or smaller than the reflection time of the pipe line are advantageous. It will be seen later that the law of closure of the main valve is of no appreciable importance, and only the time of closure must be kept within certain limits. Therefore, the control mechanism of the main valve may be of extremely simplified design.

Under the most severe operating conditions of the emergency valve, the pressure at the valve drops instantaneously to atmospheric and the velocity of the water increases considerably beyond the value of normal operation. The subnormal surge wave thus created, being reflected with negative sign at the reservoir, returns to the valve as a supernormal wave and, in order to limit the pressure rise to the specified maximum value, the wave must find the by-pass valve open to limit the reflection of the wave[6] and to act as a surge suppressor. The degree of initial opening and, therefore, the size of the by-pass equipment is definitely determined by the maximum permissible surge. The closing rate for the by-pass valve is determined by the necessity of limiting the discharge after rupture to a minimum. The pressure at the valve is held as nearly constant as possible to reduce the velocity of the water column in the minimum time.

The automatic by-pass valve may be of the needle, gate, plunger, or butterfly type and may be used as a conventional hand-operated by-pass valve for filling the line beyond the main valve to permit opening of the main valve under balanced pressure conditions. The size of the by-pass valve being considerably smaller than that of the main valve, the forces acting on it are greatly reduced and the mechanism controlling time and rate of closure can be made more rugged without excessive cost.

The proper combination of quick-closing main valve and slower-closing by-pass valve results in limiting the maximum surge and securing a minimum of discharge after rupture.

Throttling Function

In the following discussion, symbols with the subscript 1 refer to the by-pass piping and those with the subscript 2 refer to the by-pass valve. Symbols without subscripts, except that for time, refer to the main pipe line above the valve shown in Fig. 1. The symbols used are those recommended by the Water-Hammer Committee of the A.S.M.E., except when otherwise stated.

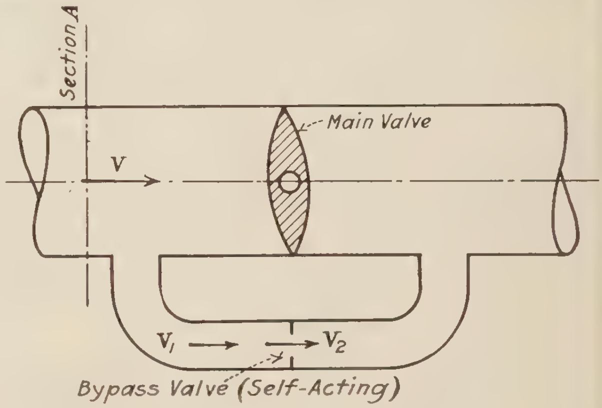


Fig. 1 Arrangement of Shutoff Valve

The pressure head H_t, necessary to force water with the velocity V_t, through the by-pass, is given by the elementary relation

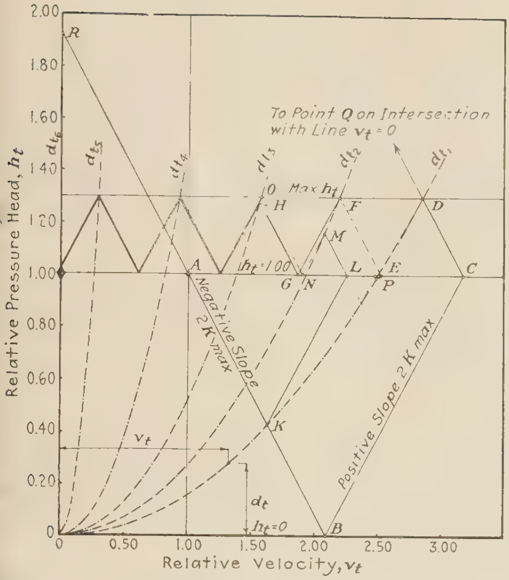
$$H_t = \left(\frac{A}{A_1}\right)^2 \left(\frac{V_t^2}{2g}\right) \left[1 + \zeta_e + \lambda \frac{L_1}{D_1} + 2\zeta_b + \alpha \left(\frac{A_1}{A_2} - 1\right)^2\right]$$

$$= \beta_1 \frac{V_t^2}{2g} = f(V_t, t) \ldots\ldots [1]$$

The terms in brackets are made up of the velocity head, entrance loss, pipe friction, losses in bends, and head loss in the by-pass valve. With the introduction of "relative" values, Equation [1] gives

[5] Occasionally a return surge may give an excess pressure above normal, but this special case may usually be neglected. See Étude sur les Maxima de Surpression dans les Phémomènes de Coups de Bélier, by M. Gariel, *Revue Générale de l'Électricité*, vol. 4, October 5, 1918, pp. 483.

[6] "High-Head Penstock Design," by A. W. K. Billings, O. H. Dodkin, F. Knapp, and A. Santos, Jr., formulas [8] and [10], p. 38; and Fig. 4, p. 56. See also discussion of this paper by E. B. Strowger, formulas [17] and [18], p. 136. Symposium on Water Hammer, The American Society of Mechanical Engineers, 29 West 39th Street, New York, N. Y., 1933.



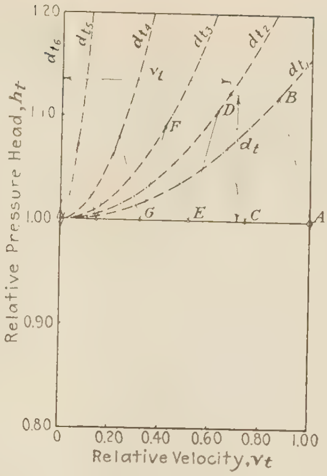


Fig. 3 Diagram for h_t-v_t-d_t for the Second Case

Fig. 2 (Left) Diagram for h_t-v_t-d_t for the First Case

$$h_t = \frac{H_t}{H_0} = \frac{\beta_1}{H_0}\frac{V_0^2}{V_0^2}\frac{V_t^2}{2g} = \beta_1\frac{V_0^2}{2gH_0}v_t^2 = \beta_2 v_t^2 \ldots\ldots [2]$$

where

$$\beta_2 = \frac{V_0^2}{2gH_0}\left(\frac{A}{A_1}\right)^2\left[1 + \zeta_e + \lambda\frac{L_1}{D_1} + 2\zeta_b + \alpha\left(\frac{A_1}{A_2} - 1\right)^2\right] \ldots [3]$$

The size of the by-pass piping and valve is determined by Equations [1] and [2]. Equation [3] may be plotted as a function of the diameter of the by-pass piping and the position of throttling valve.

Graphical Determination of Operation Characteristics

The procedure to be followed for the determination of the characteristics of the proposed automatic main and by-pass valve for the two limiting cases is shown in Figs. 2 and 3. In accordance with the requirement, to limit the discharge after rupture to a minimum, it is assumed that the main valve closes in a time equal to the reflection time of the pipe line. For the sake of simplicity, the friction losses between the reservoir and the safety valve have been neglected; however, these should be taken into account if of appreciable magnitude. Complete reflection of the pressure waves at the reservoir is considered.

1 *The Pressure at the Valve Drops Instantaneously to Atmospheric Pressure.* This case is defined by a complete rupture of the pipe just downstream from the shutoff valve. Operation of the pipe line before rupture is represented on the v_t-h_t diagram in Fig. 2 by the point A with the coordinates[7] $h_{t0} = 1.0$ and $v_0 = 1.0$. The relation between change of pressure and velocity at the valve is given by the simultaneous equation

$$h_{At} - h_{E(t-\mu/2)} = -2K_{max}\,(v_{At} - v_{E(t-\mu/2)}) \ldots\ldots [4]$$

Subscripts A and E refer to the pipe sections just upstream from the valve and at the reservoir, respectively, as shown in Fig. 4. In the diagram and for the first interval, Equation [4] is represented by the line AC with the negative slope equal to $2K_{max}$ and starting at point A. Shortly after rupture, the conditions at the valve are defined by $h_{t0} = 0$, thus giving point C with a considerably increased velocity of a magnitude

$$v_{t0} = \frac{2K_{max} + 1}{2K_{max}} \ldots . [5]$$

as compared with the steady velocity $v_0 = 1.0$ before rupture. This subnormal pressure wave travels up the pipe line and is completely reflected at the reservoir. The pressure and velocity conditions at section E are expressed by the simultaneous equation

$$h_{Et} - h_{A(t-\mu/2)} = +2K_{max}\,(v_{Et} - v_{A(t-\mu/2)}) \ldots . [6]$$

represented in Fig. 2 by the line BC with the positive slope $2K_{max}$. The pressure at the reservoir remaining constant, the point of intersection C of line BC with the horizontal $h_t = 1.0$ furnishes the conditions at the reservoir from the time $\mu/2$ up to the time $3\mu/2$.

But, at the time $t_0 = 0$, the main valve begins to close, and at the end of the first interval, at time $t_1 = \mu$, it is completely shut. Now the water begins to flow through the by-pass. Plotting the throttling function d_t, as shown in Fig. 2, the pressure rise caused by closing of the main valve and at time t_1 is obtained as point of intersection K with line AB and parabola d_{t1} in accordance with Equation [4]. It is now possible to determine that value of the throttling function which keeps the total relative pressure within the specified limit $h_{t max}$. This maximum pressure is determined by the initial subnormal surge wave, reflected at the reservoir with negative sign and arriving at the valve at time t, as a supernormal wave. At the valve and at the time t_1, Equation [4] again holds and gives point D as intersection of the line CD with the horizontal $h_{t_{max}}$. Through this point D must pass the characteristic d_t of the by-pass, thus furnishing the coefficient β_2 which as a value of

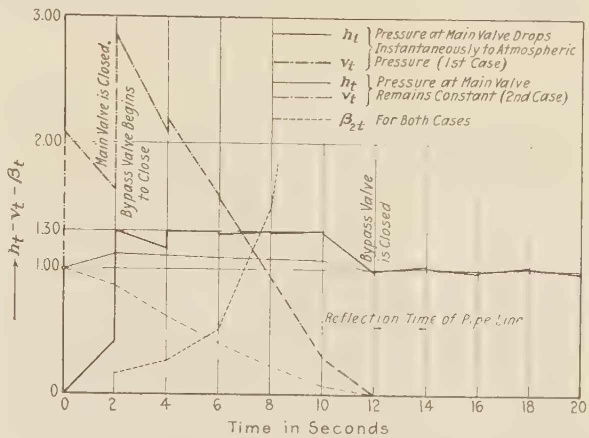


Fig. 4 Curves for Surge Versus Time, Velocity Versus Time and Opening of the By-Pass Valve for the Two Limiting Cases of Shutoff-Valve Operation

[7] The symbol h_t is the total relative pressure head equal to H_t/H_0 instead of Z^2 as recommended by the Water-Hammer Committee of the A.S.M.E. In connections with the simultaneous equations the symbol h_t is more convenient.

$$\beta_{2t1} = \frac{(1 - \Delta h)\, 4K_{\max}^2}{(2K_{\max} + 2 - \Delta h)^2} \qquad [7]$$

where Δh is the permissible maximum relative pressure rise above static head. It is interesting to note that at the end of the first interval two simultaneous values of pressure rise occur, represented by the points K and D in Fig. 2 and caused by the main-valve closure and reflection of the primary surge drop at the reservoir, respectively.

At time t_1 the by-pass valve begins to close. As the valve closing takes place in such a manner that the pressure rise is practically constant in order to decelerate the water column as rapidly as possible, further application of Equations [4] and [5] gives the points F, M, O, and H in Fig. 2 for the surge at the end of the second and third intervals. At these times, the maximum surge at the valve is determined either by the initial surge drop (point F) or by the pressure rise set up by the main-valve closure (point O). Points F and O, of course, are located by the condition of maximum surge at the valve. The surges at intervals of the reflection time may be determined in a similar manner.

The closing time of the by-pass valve is determined automatically by the construction just described and depends on maximum permissible surge and characteristics of the pipe line.

The settings of the by-pass valve at times t_2 and t_3 are given by the equations

$$\left.\begin{aligned} \beta_{2t2} &= \frac{(1 + \Delta h)\, 4K_{\max}^2}{(2K_{\max} + 2 - 3\,\Delta h)^2} \\ \beta_{2t3} &= \frac{(1 + \Delta h)\, 4K_{\max}^2}{(2K_{\max} + 2 - 5\,\Delta h)^2} \end{aligned}\right\} \ldots\ldots\ldots [8]$$

or generally

$$\beta_{2tn} = \frac{(1 - \Delta h)\, 4K_{\max}^2}{[2K_{\max} + 2 - (2n - 1)\,\Delta h]^2}$$

where n is the number of intervals

In practice it may be necessary to adjust the values of β_2 slightly in order to conform with the characteristics of the device used for controlling the travel of the by-pass valve. Several of the devices used for this purpose comply very well with the required operation. It should be noted, however, that an appreciable departure from the theoretical law of closure causes serious disadvantages and allows a greater total discharge after rupture of the pipe line. Should the by-pass valve fail to close, the velocity in the pipe line attains a steady value of

$$v_t = \frac{2(K_{\max} + 1) - \Delta h}{\sqrt{(1 + \Delta h)}} \ldots\ldots\ldots\ldots\ldots [9]$$

as shown by point P in Fig. 2.

Closure of the by-pass and main valve in a time equal to or less than the reflection time of the line produces a surge equal to

$$\Delta h'_{\max} = 2(K_{\max} + 1) \ldots\ldots\ldots\ldots\ldots [10]$$

represented by point Q in Fig. 2.

Compared with the Joukowsky surge (point R)

$$\Delta h_{\max} = 2K_{\max}$$

this emergency condition produces a pressure rise which in all cases is considerably greater than usually assumed. The ratio of the two surges amounts to

$$\frac{\Delta h'_{\max}}{\Delta h_{\max}} = 1 + \frac{1}{K_{\max}} \ldots\ldots\ldots\ldots\ldots [11]$$

It is thus seen that for high-head installations with values of $K_{\max}$ less than unity, the conditions become especially unfavorable.

2 The Pressure at the Valve Remains Constant. This case is represented by a pipe rupture at a considerable distance from the shutoff valve. The throttling function d_t obtained for the first case is not plotted with the horizontal $h_t = 0$, but with the line $h_t = 1.0$ as base line, as shown in Fig. 3. Application of the simultaneous Equation [4] gives the point B and Equation [5] gives the point C. The resulting surges are considerably smaller than those of the first case.

The surge and velocity time curves as well as the by-pass coefficient β_2 for the two limiting cases are plotted in Fig. 4 for a pipe line with the following characteristic values

$$K_{\max} = 0.460; \; h_{t\max} = 1.30; \; \frac{2L}{a} = 2 \text{ sec}$$

It is interesting to note that for this specific example and for the first case the velocity of the water at the valve increases instantaneously from a value of $V_0 = 1.0$ for steady operation to a value of $V_{t0} = 2.08$ after rupture and only after about 4 intervals or 8 sec drops below the initial value.

The velocity in the second case does not exceed the steady value $V_t = 1.0$, but drops steadily to zero velocity. Thus, the first case with the pressure at the valve falling instantaneously to atmospheric pressure determines the design of the emergency closing valve.

Cavitation Caused by Subnormal Surge Conditions

The surge drop produced by a rupture of the pipe just below the shutoff valve travels up the pipe line unchanged, provided the wave front does not meet the zero-pressure line of the pipe. This zero line is parallel to and below the pipe line a distance equal to the atmospheric minus the vapor pressure of the water as shown in Fig. 5. With a sloped pipe line, the column of water between section X and the reservoir cavitates partially to a varying degree, but no separation of the water column occurs. Reflection at the reservoir takes place only with a small percentage of the magnitude of the primary surge drop and at the time the reflected wave arrives at section X, the water column is brought back to its original state without cavitation, producing a corresponding surge. Further investigations are necessary to clear up completely such water-hammer conditions with cavitation.

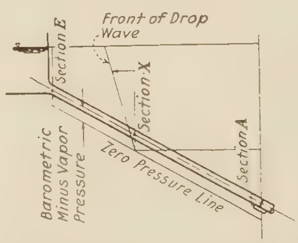


Fig. 5 Subnormal Accelerating Surge-Drop Wave Which Causes Cavitation in Pipe Lines

Conclusion

Time and rate of closure of automatic valves installed in main branches of waterworks distribution systems or in turbine penstocks can be determined in such a manner that the maximum water hammer under the worst conditions does not exceed a given limit and also that the total discharge after pipe rupture becomes a minimum. To accomplish this purpose, the main valve is provided with an automatic by-pass valve of definite dimensions. The design of the automatic-valve installation is determined by the emergency condition of pipe rupture just below the valve.

Complete Characteristics of Centrifugal Pumps and Their Use in the Prediction of Transient Behavior

By R. T. KNAPP,[1] PASADENA, CALIF.

This paper describes the technique of determining the complete operating characteristics of a hydraulic machine such as a centrifugal pump or a turbine, together with a method of presenting these characteristics in a convenient manner on a single diagram. The characteristics of a modern, high-head, high-efficiency pump are analyzed and presented in the manner proposed. The use of these complete characteristics for the prediction of the behavior of the machine during operating transients is discussed and the analytical background is presented. The assumptions involved are investigated and experimental checks of their validity are offered. The interrelationships between the hydraulic characteristics of the machine and the pipe line are indicated.

If THE possible operating conditions of hydraulic-turbine and centrifugal-pump installations are compared, it soon becomes apparent that the pumps are subject to much wider and more involved variations than are the turbines, especially during the transient states of starting, stopping, or emergency operation. In turbines the direction of flow and the direction of rotation are always the same, even in case of a breakdown of the machine itself or trouble in the penstock and auxiliary equipment. Thus the machine performance always lies in the quadrant of normal turbine operation, and since its hydraulic characteristics are very well-known in this quadrant, it is a comparatively straightforward matter to predict the complete performance during any possible transient condition. On the other hand, under similar conditions with a pump installation, the flow can completely reverse direction, as can the rotation. The machine in this case ceases to be a pump, and after passing through a zone of energy dissipation, becomes a runaway turbine. This great variation in performance gives rise to many questions, such as the runaway speed of the machine as a turbine, the time of reversal, the magnitude of the accelerating forces, the effect on the surge cycle in the discharge line, the maximum and terminal reverse rates of flow, and so on. Unfortunately these questions are very difficult to answer, because, although the hydraulic performance of the machine is well-known as long as it is acting as a pump, comparatively little study has ever been made of the performance as an energy dissipator or as a turbine.

The objective of this study has been, therefore, to explore these little-known regions of performance and to attempt to use the resulting information to answer some of these important questions.

Previous Investigations

In 1931 Kittredge and Thoma (1)[2] published an article on "Centrifugal Pumps Operated under Abnormal Conditions." This paper described experiments carried on with a small pump for the purpose of obtaining performance characteristics from which the behavior of the pump during sudden changes of operating conditions could be predicted. In these experiments the pump was operated under conditions of negative head, delivery, and speed, in addition to the normal range of performance. As an outgrowth of the work a series of investigations was undertaken in the hydraulics laboratories of the California Institute of Technology, under the direction of the author.

In the fall of 1931, Boothe and Lewis (2) started a preliminary investigation on a $1^1/_2 \times 10$-in. single suction pump. Although the results were very interesting, it was felt that the pump was too small and the efficiency too low for them to be completely typical of modern installations.

In the spring of 1932, two 4-in. high-head high-efficiency pumps were made available through the generosity of the Byron-Jackson Company. These pumps were installed in the summer of 1932 and work was carried on with them for about two years. The first results were outlined by Haynes and Sauermann (3) in 1933, and in 1934 a more complete presentation was made by the present author (4). The study also furnishes most of the background for this article.

Determination of Complete Characteristics

(A) *Laboratory Equipment.* The objectives of the program demanded that the pump under test be so installed that it could be operated under all possible conditions of flow, head, and speed, both as a pump and as a turbine. Therefore, of the two available, the one with the lowest head and capacity was selected as the test pump, while the other was designated as the service or supply pump and was connected so that it could deliver either to the suction or the discharge lines of the test pump. The test pump itself was connected to a Sprague electric dynamometer which was capable of being operated either as a motor or a generator at any speed up to 3500 rpm and in either direction of

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Contributed by the Hydraulic Division for presentation at the Second Water-Hammer Symposium, in cooperation with the American Society of Civil Engineers and the American Water Works Association, at the Annual Meeting of The American Society of Mechanical Engineers, to be held in New York, N. Y., Dec. 6 to 10, 1937.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until March 10, 1938, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

rotation. It had a capacity of 100 hp or about twice the maximum required by any operating condition of the pump. Fig. 1 is a diagram of the complete equipment. It will be noted that a spray pond and auxiliary pump are provided for cooling the system. This was necessary because the combined power input of the test and service pumps went as high as 150 hp, and since the system was a closed one with a comparatively small volumetric capacity, such a power input would have caused a rapid rise in temperatures if arrangements had not been made to dissipate it.

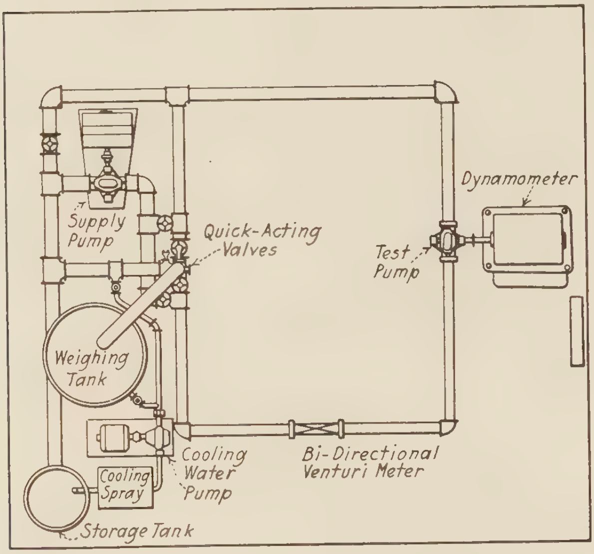

Fig. 1 Pipe Layout for Pump Tests

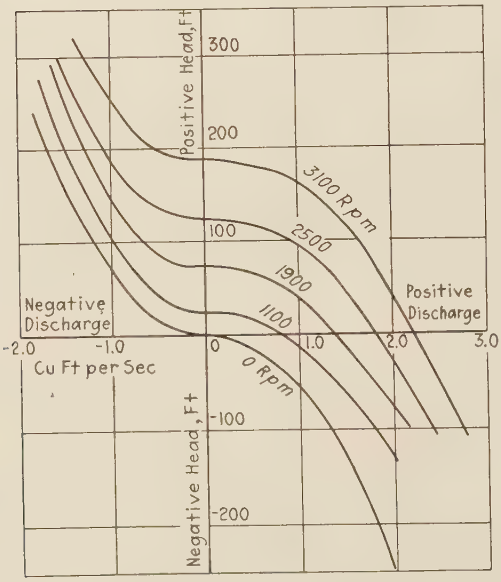

Fig. 2 Positive-Rotation Head-Discharge Curves

(*B*) *Instruments.* The basic method of measuring the rate of flow was by use of the weighing tank and synchronized chronograph. The working secondary standard was a bidirectional venturi tube, calibrated in place. This tube was constructed in the laboratory, and consisted of two symmetrical long-taper cones with piezometer rings in the center and at each end. Speed was measured with a magnetic-clutch revolution counter or by a revolving contact which recorded each tenth revolution on the chronograph drum. Low pressures were measured by means of a six-foot mercury manometer, while special ground-piston, fluid-pressure scales reading to $^1/_{10}$ lb per sq in. were constructed for the high-pressure measurement. Torque was measured directly by the scales on the cradle dynamometer. At all points care was taken to insure the accuracy of the operations.

(*C*) *Experiments.* The schedule of tests consisted of a series of constant-speed runs, both in the normal and in the reverse direction. For each speed the discharge was varied from a maximum-negative to a maximum-positive value. By negative discharge it is meant that the flow was from the discharge to the inlet of the pump. The range of discharges was about the same for all speeds, varying from about —200 to + 150 per cent of the normal for the pump when operating at 3100 rpm. It should be

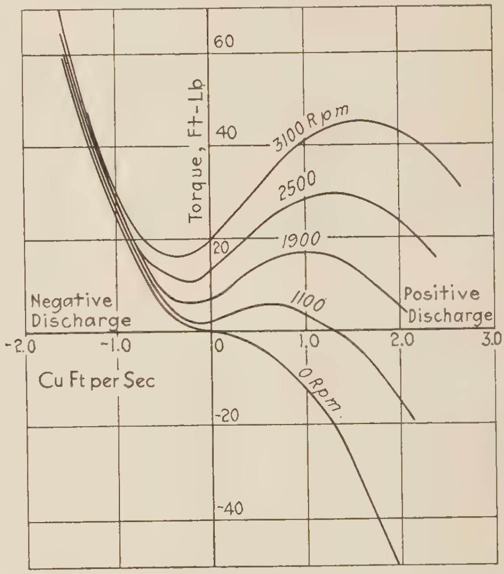

Fig. 3 Positive-Rotation Torque-Discharge Curves

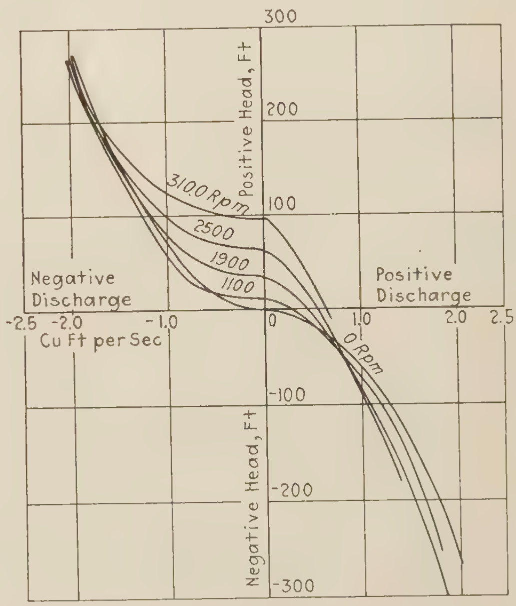

Fig. 4 Negative Rotation Head-Discharge Curves

noted that one of the constant-speed runs in the series was that of zero speed. For this run the impeller and shaft were locked to a torque arm and torque readings taken directly as the flow was varied over the usual range.

(*D*) *Constant-Speed Curves.* The results of these runs were first plotted in the series of constant-speed curves shown in Figs. 2 to 5 inclusive. Figs. 2 and 4 are the head-discharge curves for positive and negative rotation, while Figs. 3 and 5 show the torque-discharge relations. If these curves are examined, several interesting points will be noted; for example, (*a*) there are no discontinuities even at points where head, discharge, or torque reverses sign, (*b*) the zero-speed curve is smooth and has the characteristic shape of the family.

(*E*) *Complete Characteristics Diagram.* In the papers of Kittredge and Thoma (1), previously referred to, the final results were presented in two series of curve sheets. The first set showed the variations in head and power plotted against the discharge, while the speed was held at either a constant positive or negative value. The second set plotted the same variables against speed while the discharge was held at given constant values. The first set was of course analogous to Figs. 2 to 5 of this article.

When the final presentation of the results of the current test was considered, it was suggested by Prof. Th. von Kármán of

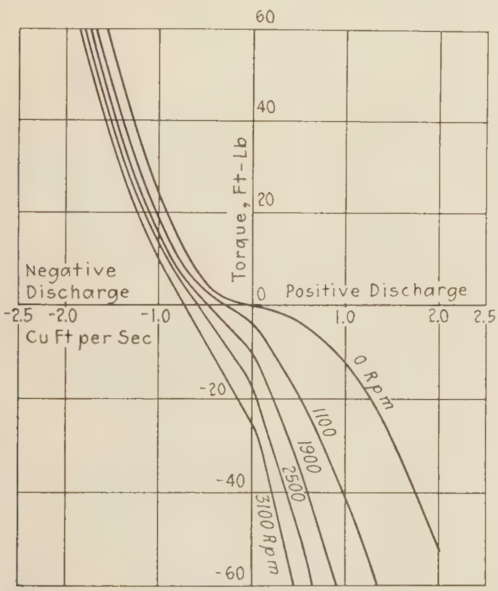

FIG. 5 NEGATIVE-ROTATION TORQUE-DISCHARGE CURVES

the Institute that a more comprehensive and useful picture of the complete performance of the machine would be obtained if all of the results were presented on a single four-quadrant diagram having as coordinates the discharge and the speed. The merits of this particular diagram are more easily seen if a brief study of its characteristics is made before it is used. Fig. 6 is an explanatory chart for such a diagram, and shows the way the various zones of operation separate themselves. It will be seen that there are two zones of pump operation, the normal one in the first quadrant and the abnormal or reverse rotation one in the second quadrant. Likewise there are two zones of turbine operation, the normal one in the third quadrant, and the abnormal or outward-flow one in the first quadrant. Each of these zones of possible useful operation is separated from the others by zones of energy dissipation in which no useful work is done either on or by the fluid. For example, the entire fourth quadrant is such a zone, and separates the zone of normal pump operation from that of normal turbine operation. Since all possible conditions of operation may be represented on this chart it may be named the "complete characteristics diagram" of the hydraulic machine tested. It is of course equally applicable either to a pump or a turbine.

Fig. 7 is the complete characteristics diagram of the 4-in. double-suction pump tested. The data are presented as series of

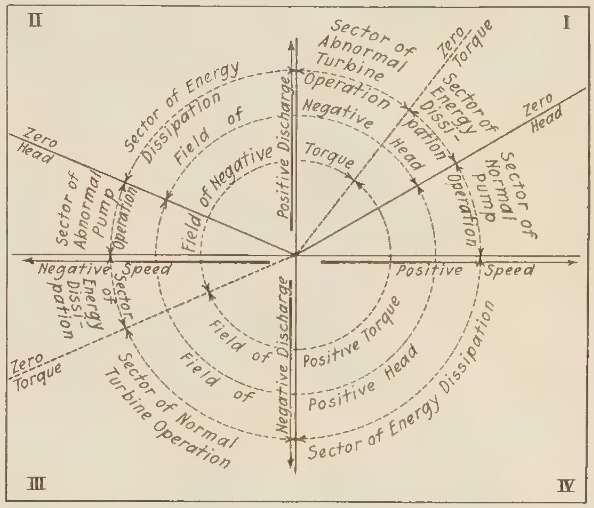

FIG. 6 EXPLANATORY CHART FOR COMPLETE CHARACTERISTICS DIAGRAM

contour curves of constant values of head and torque, the full lines being lines of constant head and the dotted ones lines of constant torque. It is interesting to observe that both pairs of zero-head and zero-torque lines are practically straight. They, of course, form the asymptotes for the corresponding families of contour curves. Each family shows reasonable agreement with the similarity laws, i.e., the entire family can be approximately calculated from any single curve. There are some deviations, however, in the zones of abnormal operation and energy dissipation. These deviations are probably caused principally by incipient or developed cavitation in some part of the machine. Therefore, these zones of the diagram are somewhat sensitive to the value chosen for the base pressure for the particular condition of flow being tested. From this point of view it would appear advisable to standardize on the normal pressure existing at the pump inlet under the condition of the proposed installation, if the diagram is to be used for calculating transient behavior.

For some purposes it is desirable to add a third set of contours to this diagram, i.e., lines of constant efficiency. They would exist only in the four zones of useful operation, since the efficiency is zero throughout all four zones of energy dissipation. In the abnormal pump and turbine zones all efficiencies are very low, but high efficiencies of the same order of magnitude were found in both the normal pump and turbine zones.

Many of the critical operation points can be read directly from the diagram. For example, for one given head the speed at which pumping ceases and flow reverses is given by the intersection of that constant-head line with the speed axis, which is the line of zero flow; the negative flow at which the machine reverses its direction is determined by the intersection of the same constant-head line with the discharge axis, which is the line of zero speed; and the corresponding turbine runaway speed is located by the crossing of the zero-torque line by this same constant-head con-

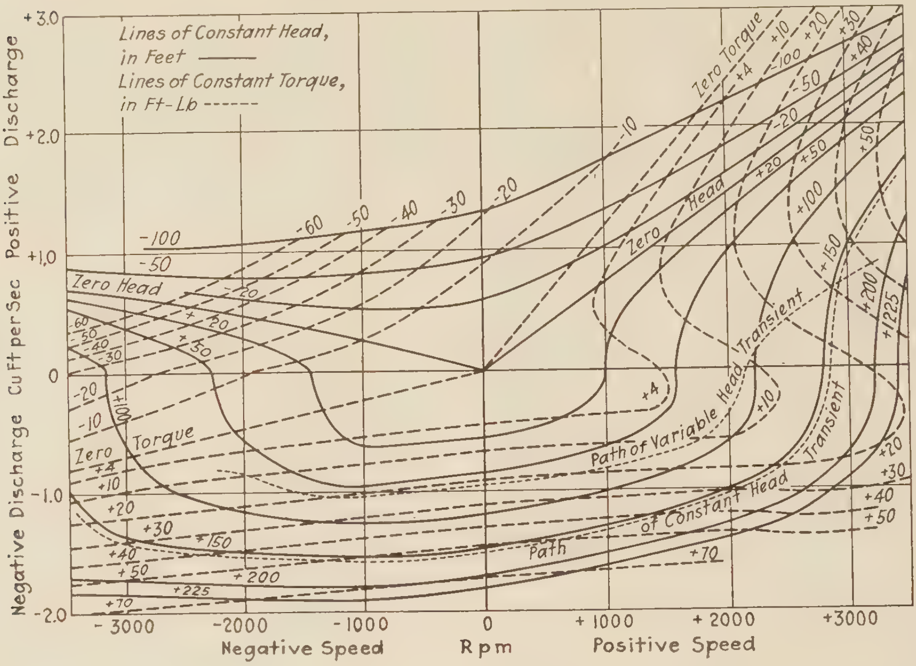

Fig. 7 Complete Characteristics Diagram for 4-In. Double-Suction Pump

tour. The maximum reverse flow and the corresponding speed are determined by the point of tangency of a horizontal line with this head contour.

Prediction of Transient Behavior

(*A*) *Basic Principle.* As has been previously stated, the complete characteristics diagram, Fig. 7, covers all possible conditions of operation of the machine within the ranges of speed and discharge shown. Therefore, if there is a change in the operating conditions it must be possible to plot the path of this change on the diagram. Consider a simple example: Suppose the test pump operated at 3200 rpm and discharged water into a reservoir 150 ft above the suction supply, through a submerged outlet pipe line so large that both friction losses and possible surges could be considered negligible. Then, no matter what the pump did the head would remain constant. Thus the operating point would always remain on the 150-ft head line on the diagram. If the driving power were removed, the operating point would move down the line out of the first quadrant, through the fourth and into the third quadrant until it intersected the line of zero torque at a negative speed of approximately 3600 rpm (turbine runaway) which would be the new point of equilibrium. Simultaneous values of discharge, speed, and torque can be read from the diagram for every point along the constant-head line. Since the driving torque was removed, the values of torque indicated by the diagram must be supplied by the deceleration of the rotating mass of the system. If the polar moment of inertia of this rotating system is known, it is possible to calculate the time-speed relations existing during the change. If any other speed-head path had been followed, similar calculations could have been made.

The usefulness of this method is that it offers a way of calculating the time characteristics of such events as normal starting and stopping of a pumping plant, abnormal shutdown due to power failure, the behavior of the pump in case of shaft breakage, or other emergency conditions arising in plant operation. Of course, it alone does not give the entire picture, for in the normal installation the surge pressures in the discharge line during transients are of major inportance. However, the complete characteristics diagram of the machine furnishes the information needed to calculate the surge, and the surge pressures are necessary in order to plot the speed-head path on the diagram. This apparently indicates that a step-by-step calculation would be necessary for the complete determination of the performance. The results should be well worth the effort, for, with the information they would furnish, the designer would be in position, for example, to determine with surety whether or not relief or quick-closing valves would be necessary to the operation of the system, and if so their complete time characteristics.

By the use of the graphical method for water-hammer problems described by Schnyder (6), Bergeron (7), and Angus (8), this step-by-step solution can be greatly simplified when studying surge conditions in pump-discharge lines.

(*B*) *Assumptions Involved.* The use just proposed for the complete characteristics diagram involves major assumptions that must be thoroughly understood and borne in mind during the application. The principal assumption upon which most of the others depend, is that the instantaneous performance of the machine for any given set of momentary conditions occurring during a transient is identical with the steady-state performance for those same operating conditions. In this assumption two others are implied, (*a*) that two or more types of flow cannot exist within the machine for one given set of operating conditions, even momentarily, and (*b*) that the momentary accelerating forces exerted on the fluid *within the machine* during the transient are small in comparison to the forces required for normal steady operation at that particular state. If these assumptions do not hold true within the limits of accuracy desired for the calculations or if corrections cannot readily be applied to bring them within these limits then the method is not applicable. It was, therefore, decided to try to carry out an experimental test of the method and the assumption for some simple cases of transient behavior that could be studied within the limitations of the laboratory equipment. However, before these experiments are

described, the analytical background of the prediction of transient behavior will be discussed in more detail.

(*C*) *Analytical Background.* The equation of motion of a rotating system is,

$$T = I\frac{d\omega}{dt} \qquad [1]$$

where

T = unbalanced torque applied to system
I = moment of inertia of the system about the axis of rotation
ω = angular velocity
t = time

if ω be replaced by its equivalent $\frac{2\pi N}{60}$ this becomes

$$T = \frac{\pi}{30} I \frac{dN}{dt} \qquad [2]$$

where N = revolutions per minute.

This may be integrated to obtain

$$t_2 - t_1 = \frac{\pi}{30} I \int_{N_1}^{N_2} \frac{1}{T}\, dN \qquad [3]$$

If the torque is known as a mathematical function of speed it may be possible to integrate this analytically, but if the relation is only known empirically it is always possible to evaluate the integral graphically. The latter is the more normal case. The procedure is simply to plot $\frac{\pi I}{30T}$ as a function of N. The area under the curve between any two values of speed, N_1 and N_2, is equal to the time required for the speed to change from N_1 to N_2. If the entire integral curve is plotted it will give the speed as a function of the time.

The method just described is precisely that used to obtain a prediction of the time-speed relations during a pump transient from the known speed-head relations and the complete characteristics diagram. Of course, as mentioned under section (*A*) of this division of the article, the speed-head relationship is often an implicit function of the discharge-line characteristics and the machine characteristics. In such a case it would probably be necessary to plot and integrate the differential curve step by step. After the integral curve has been plotted, giving the speed-time relationship, one more piece of information can be obtained. The speed values of this curve are identical with those on the complete characteristics diagram, following the line of the known speed-head relation. But along this line, each speed corresponds to a definite value of the discharge. Consequently values from the two diagrams can be combined to give the discharge-time curve for the transient change.

(*D*) *Experimental Verification of Transient Predictions.* A study of the method of transient analysis just presented suggested the most direct type of experimental check. This was to carry the pump through a known change of operating conditions, experimentally determining the actual speed-time relationship, and comparing this with the prediction calculated according to the method described from the observed speed-head relationship. If the two agreed, this would be direct evidence of the validity of the major assumption of the method.

The simplest way of carrying on these experiments appeared to be to start with the test and service pumps both operating and discharging into a common line. If the power supply of the test pump was cut off then the supply pump would furnish the pressure necessary to maintain the head. The test pump would decelerate, reverse, and come up to speed as a runaway turbine.

The regular instrumentation permitted such experiments. The primary readings necessary were the speeds and times. These were automatically recorded on the chronograph record of time vs. total revolutions. The instantaneous speed was determined accurately by measuring the number of revolutions recorded in a small interval of time. Two types of runs were made. By controlling the service-pump discharge valve the head on the test pump was kept constant independent of the test-pump speed. By the other method the setting of the discharge valve was not changed, and, as the test pump slowed down, the discharge pres-

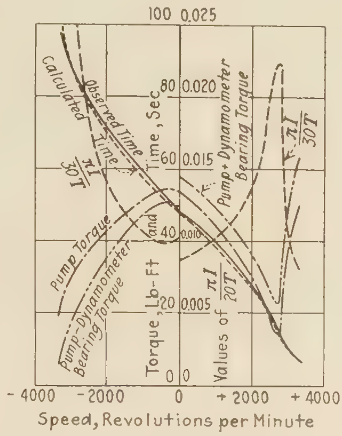

Fig. 8 Transient Speed-Time Relations
(Constant-head run.)

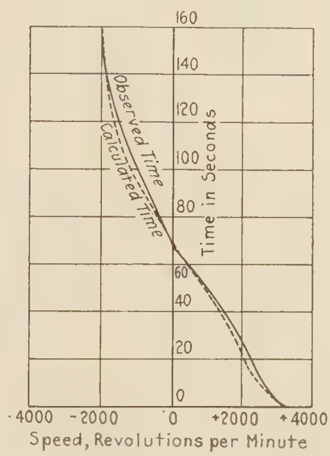

Fig. 9 Transient Speed-Time Relations
(Variable-head run.)

sure fell until the equilibrium condition was reached where the test pump ran as a runaway turbine on the water supplied by the service pump. For this case the speed-head data were simultaneously recorded by observers.

The method of predicting the transient relationship was the same for both tests. First the head-speed path was plotted on the complete characteristics diagram directly from the readings taken during the experiment. From this path the torque-speed relationship was plotted. During the actual run an additional torque acted on the rotating system. This was the dynamometer bearing friction, the value of which had previously been carefully determined. This friction always opposed the direction of mo-

tion. The action of the water opposed the motion until the runner had been brought to a standstill, then produced an acceleration until the runaway speed was reached. The resultant torque on the system was thus the sum of the two from the initial point to the point of zero speed, and the difference of them from that point to the point of equilibrium runaway condition.

Fig. 8 shows a constant-head run. The speed-head path may be seen in light dotted lines in Fig. 7. The curve of Fig. 8 labeled pump torque shows the speed-torque readings from Fig. 7. The curve labeled pump and dynamometer torque shows the addition of the dynamometer bearing friction. The reciprocal of this curve, multiplied by the constant $\frac{\pi}{30} I$, was then plotted. The integral curve of this is the dotted speed-time curve. The solid speed-time curve shows the actual experimental values.

Fig. 9 shows the predicted and observed speed-time curves for a case where the head varied during the transient. The path of this run may also be seen in light dotted lines in Fig. 7. These curves are typical of the group of check runs that were made. In all cases the agreement between the predicted relationship and the observed values was better than could reasonably be expected.

In one respect, these check runs do not completely satisfy all questions. Due to the large moment of inertia of the dynamometer armature, all the transient runs which it was possible to make have a relatively slow rate. In the faster one shown, Fig. 8, it is seen that it took about 48 sec until the point of reversal of rotation, and about 100 sec until the equilibrium runaway speed was approached. This is considerably longer than would be the case with the average field installation. On the other hand, the agreement here is so good that a considerable additional error would be tolerable before the usefulness of the results would become impaired.

Continuation of Investigation

During the spring of 1934, while the last of these tests were being completed, a new hydraulic-machinery laboratory was being designed and constructed at the California Institute of Technology, in cooperation with the Metropolitan Water District of Southern California. It was put into operation in August of that year and since that time an intensive program of study of the problems of high-head pumping plants of the Colorado River Aqueduct has been carried on. A description of this laboratory by the author (5) may be found in the Transactions. A part of this work has been the determination of the complete characteristics of several of the model pumps tested in the laboratory. Fig. 10 shows one example of these diagrams which is published through the courtesy of the Byron-Jackson Company and the Metropolitan Water District. The pump, the characteristics of which are shown, is a much larger one than that used in the foregoing work, requiring about 300 hp when operating at the point of best efficiency against a head of 300 ft. It represents the best current pump practice, since in the model it showed a test efficiency of over 92 per cent. It is anticipated that in the near future

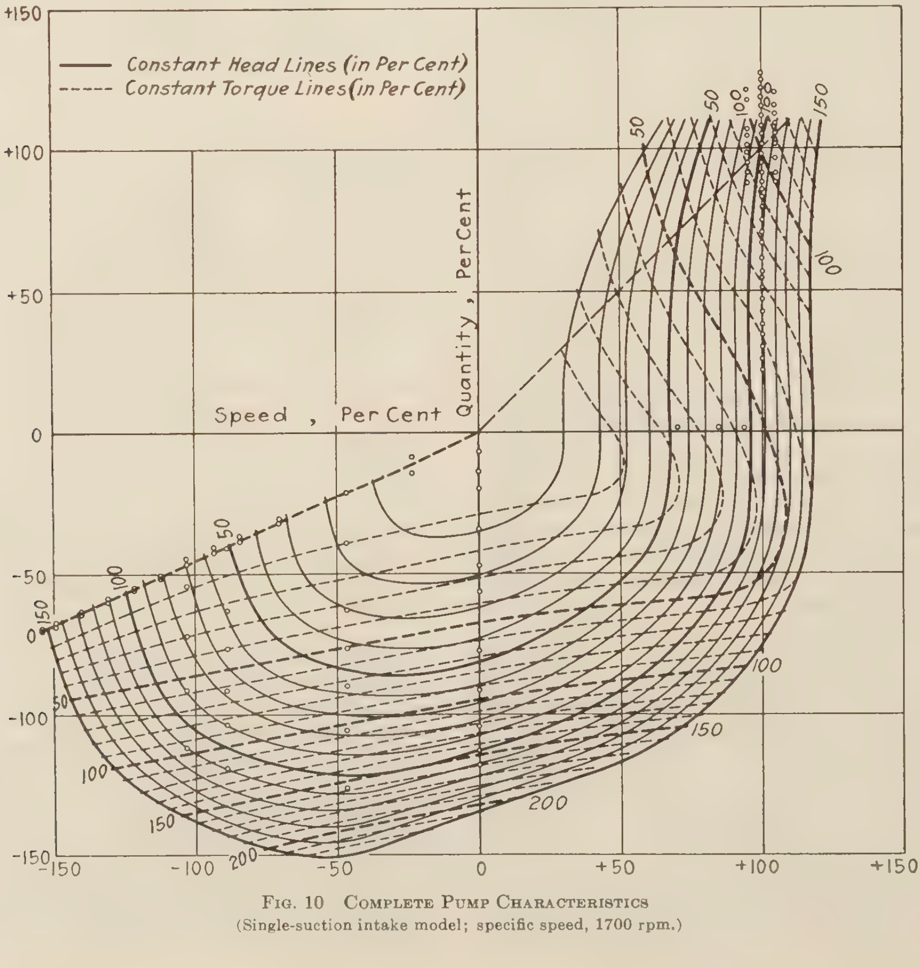

Fig. 10 Complete Pump Characteristics
(Single-suction intake model; specific speed, 1700 rpm.)

more complete tests will be made of this and other large-scale models of varying specific speeds, and also that other transient check runs at higher accelerations can be carried out to test more thoroughly the validity of this method of calculating transients.

Acknowledgment

In addition to the men whose names are mentioned in references (2) and (3) and to the Byron-Jackson Company and the Metropolitan Water District, the author wishes especially to acknowledge the assistance of Dr. Geo. F. Wislicenus, who worked diligently with him in installing the apparatus and instruments in the original laboratory, who helped direct the experimental work, and who collaborated with him on the analysis and application of the results.

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Experiments and Calculations on the Resurge Phase of Water Hammer

By JOSEPH N. LECONTE,[1] BERKELEY, CALIF.

The author develops a theoretical expression for the multiple recoil or resurge effect of a long water column following the instantaneous, or approximately instantaneous, closing of a valve at the lower end of the column. The time period between successive resurges or hammer blows is calculated by the developed expression and is compared with time periods recorded from tests by an oscillograph. The computed and observed results are presented in tabular and graphical form for several of the tests.

IN THIS paper an attempt has been made to develop a theoretical expression for the multiple recoil or resurge effect of a long water column following the instantaneous (or approximately instantaneous) closure of a valve at its lower end. It is a matter of ordinary experience that after the compression wave in such a pipe has completed the round trip from the valve to the reservoir and back, the entire column may recoil from the valve, leaving a vacuous space behind, and then return to the valve, making a second hammer blow similar to, though less violent than the first. In the same way this resurge may repeat itself many successive times until the original energy of the mass has been exhausted. These repeated blows are of continually diminishing intensity or amplitude, and the time period between successive blows is a continually diminishing one. It is to investigate the time element that the following work has been planned.

In order to obtain a physical conception of what probably takes place, consider the ideal condition shown in Fig. 1. Imagine water flowing through a long pipe from a reservoir A under constant head h to have reached its terminal velocity v_1. When the valve at B is suddenly closed the wave front travels backward according to the well-known law, and when at C we may imagine the water from C to B to be entirely at rest under pressure p in an expanded pipe, while that from D to C is still advancing with velocity v, approximately the same as v_1. When the wave front reaches D the entire pipe is full of water at rest and under stress. The reverse phase then takes place, and by the time the wave front returns to B, the entire mass is flowing in the reverse direction with some velocity v. Since the column cannot be brought instantly to rest, it pulls away from B leaving a vacuum behind it, and continues to move until its kinetic energy has been exhausted in overcoming friction, overcoming the head h, and that of the atmosphere h_0. From this position of rest S, it again returns, under the influence of h and h_0, strikes the valve again, and the phenomenon is repeated continually until the surge dies out. Such a series of surges is shown graphically in Fig. 2, wherein LM is the first hammer blow, MN is the time t for the compression wave to make its round trip,

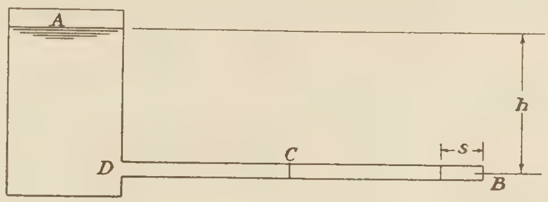

FIG. 1

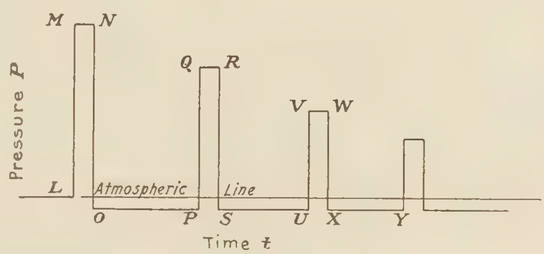

FIG. 2

OP is the time during which the column surges back and forth under vacuum, and QR is the time for the next round trip of the wave, which is the same as MN.

The results of laboratory experiment make it clear that the velocity with which the water strikes the valve on any one of these surges is always greater than that with which it starts away on the next surge. In other words the velocity at Q is greater than that at R. This relation can be expressed as

$$v_0 = Cv_{01} \quad \text{[1]}$$

v_0 being the velocity of return at R, and v_{01} that of approach at Q. The coefficient C, which seems to be of the nature of a "restitution coefficient" in the mechanics of impact and collision, is nearly constant for the set of waves of one experiment, and approximately so for different velocities in a single pipe line. Strictly it should be a function of the velocity and pipe diameter, and probably of the wall thickness and material of the pipe. It seems to diminish in numerical value as the velocity increases.

There is also a reduction in velocity due to pipe friction, as the mass slides through the pipe and back, and a destruction of velocity head as the column enters the reservoir. If we regard the pipe friction factor f, as being of mean constant value, this

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

loss can be computed as follows: Let v_0 be the initial velocity of the resurge, and v any value of the velocity thereafter, then if l is the length and d the diameter of the pipe, the force equation takes the form

$$a\gamma(h + h_0) + a\gamma f \frac{l}{d}\frac{v^2}{2g} = \frac{la\gamma}{g}\frac{dv}{dt} \qquad [2]$$

The integration of Equation [2] leads to

$$t_1 = \frac{lv''}{g(h + h_0)} \tan^{-1} \frac{v_0}{v''} \qquad [3]$$

where t_1 is the time for the column to come completely to rest, and v'' is an abbreviation for the expression

$$\sqrt{\left[\frac{2g(h + h_0)}{f(l/d)}\right]}$$

If we imagine the entire column to hold together, the distance S moved through on the rebound, while the velocity changes from v_0 to zero, will be found by integrating Equation [2] in the form

$$a\gamma(h + h_0) + a\gamma f \frac{l}{d}\frac{v^2}{2g} = \frac{la\gamma}{g}\frac{v\,dv}{ds} \qquad [4]$$

This yields

$$S = \frac{lv''^2}{g(h + h_0)}\, {}^1/_2 \log_e \left[1 + \left(\frac{v_0}{v''}\right)^2\right] \qquad [5]$$

On the return of the column we have accelerated instead of retarded flow, and the force equation is

$$a\gamma(h + h_0) - a\gamma\left(1 + f\frac{l}{d}\right)\frac{v^2}{2g} = \frac{la\gamma}{g}\frac{dv}{dt} \qquad [6]$$

The integration of Equation [6] gives

$$t = \frac{lv'}{g(h + h_0)} \tanh^{-1} \frac{v}{v'} \qquad [7]$$

where v' is an abbreviation for the expression

$$\sqrt{\left[\frac{2g(h + h_0)}{1 + f(l/d)}\right]}$$

Equation [7] can be put in the form

$$v = v' \tanh \frac{g(h + h_0)t}{lv'} = \frac{ds}{dt} \qquad [8]$$

and we have

$$ds = v' \tanh \frac{g(h + h_0)}{lv'}\, t\, dt \qquad [9]$$

or

$$S = \frac{lv'^2}{g(h + h_0)} \log_e \cosh \frac{g(h + h_0)t}{lv'} \qquad [10]$$

If we equate the two values of S, as given by Equation [5] and [10], and put t_2 for the time of return, we have

$$\log_e \cosh \frac{g(h + h_0)}{lv'}\, t_2 = {}^1/_2 \left(\frac{v''}{v'}\right)^2 \log_e \left[1 + \left(\frac{v_0}{v''}\right)^2\right]$$
$$= n \log_e \left[1 + \left(\frac{v_0}{v''}\right)^2\right]$$

or

$$\cosh \frac{g(h + h_0)}{lv'}\, t_2 = \left[1 + \left(\frac{v_0}{v''}\right)^2\right]^n \qquad [11]$$

where, from the derivation of Equation [11]

$$n = {}^1/_2 \left(\frac{v''}{v'}\right)^2$$

Equation [11] will enable us to compute t_2, and the time for the entire cycle will be $t = t_1 + t_2$.

In order to test the foregoing theory and estimate the value of the coefficient C, two water-hammer lines were installed in the hydraulic laboratory of the University of California. These were of ordinary black pipe, of 1-in. and 2-in. nominal sizes. Due to the limited size of the laboratory, it was necessary to carry these around the walls of the room, with long radius bends in the corners. The radii of these bends were about 10 ft. The lines were supplied from a standpipe 3 ft in diameter which was arranged to overflow at a definite level for constant head. The quick-closing valve was of the rotary or plug-type and was operated by a weight falling from a height of about 8 ft. The time-pressure curve was drawn by the aid of a Westinghouse oscillograph. One of the elements recorded the pressure by variation in electrical resistance of a pile of carbon buttons subjected to the pressure. A second element recorded the time of closing the valve, while a third recorded the 60-cycle power alternations, by means of which time could be measured.

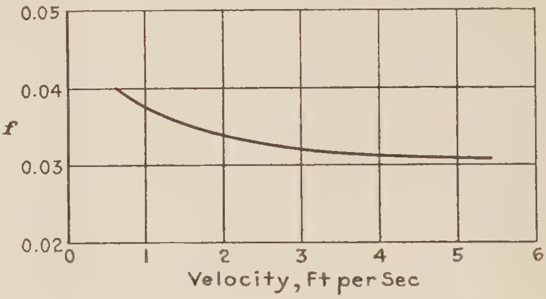

Fig. 3 Values of f for the 2-In. Pipe Line

In making a series of runs on a pipe line, the coefficient f was first obtained by running on constant head at various velocities. Such a curve for the 2-in. line is shown in Fig. 3. The pressure element was in every case calibrated by means of a dead-weight pressure tester, and a series of calibration lines were drawn across the film of the oscillograph. The plug valve was then set at a given position of throttling, the water weighed for a definite time, and then, without changing the valve, the weight was dropped and the time-pressure curve recorded. In working up the predetermined relations, the pipe-line dimensions given in Table 1 were used.

It appears that the value of the coefficient C for the first hammer blow, that is, the one caused by the closure of the valve, is slightly higher than for subsequent ones where the concussion is under vacuum. If the closure were absolutely instantaneous, perhaps no difference would be detected. The value of C is also a function of v, being smaller for higher velocities.

Sample Computation

As mentioned previously, tests were conducted with 1-in. and 2-in. pipe lines. The results of one of the tests on the 2-in. line, which was designated as test No. 5A, were computed as follows:

The initial velocity v for steady flow was 3.11 fps, the hydrostatic head h was 24.1 ft, and C was taken as 0.87. Thus, the velocity with which the water starts away from the valve is

$$v_0 = 3.11 \times 0.87 = 2.707 \text{ fps}$$

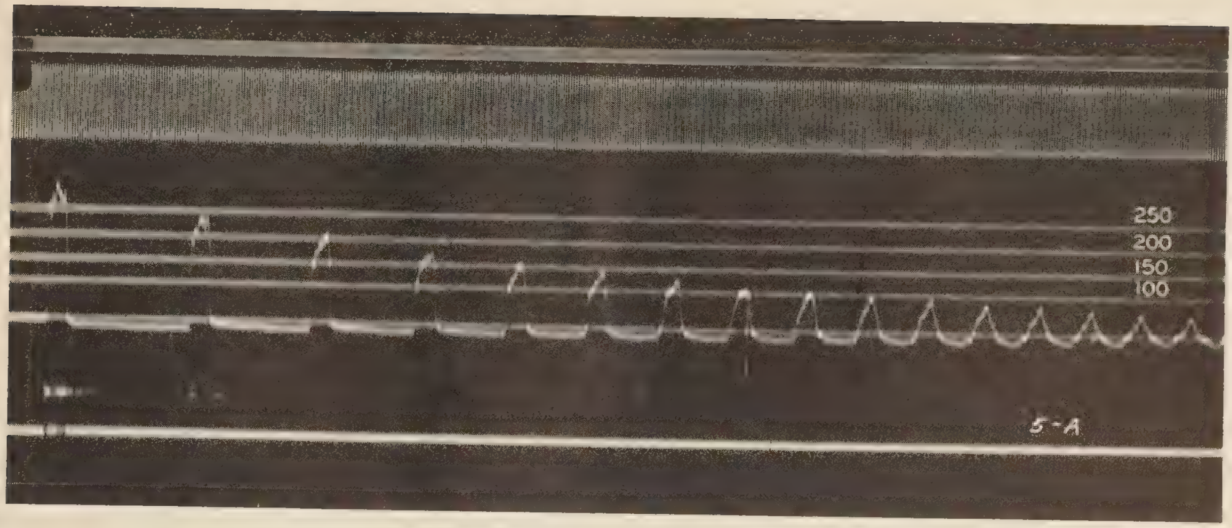

FIG. 4 OSCILLOGRAPH FOR TEST NO. 5A

For this we may take a value of $f = 0.0342$. From the foregoing values of f and h, and the pipe-line dimension given in Table 1

$$v'' = \sqrt{\left[\frac{2g(h + h_0)}{f(l/d)}\right]} = \sqrt{\left(\frac{3742}{52.15}\right)} = 8.47$$

and

$$v' = \sqrt{\left[\frac{2g(h + h_0)}{1 + f(l/d)}\right]} = \sqrt{\left(\frac{3742}{53.15}\right)} = 8.38$$

First Resurge. For the first resurge, applying Equation [3]

$$t_1 = 1.26 \tan^{-1} (2.707/8.47) = 0.39 \text{ sec}$$

and from Equation [11], where $n = 0.5(8.47/8.38)^2 = 0.51$

$$\cosh 0.802\, t_2 = [1 + (2.707/8.47)^2]^{0.51}$$

or

$$\cosh 0.802\, t_2 = 1.102^{0.51} = 1.0508$$

from which $0.802\, t_2 = 0.3175$, and $t_2 = 0.396$ sec. Therefore, $t = t_1 + t_2 = 0.786$ sec. The measurement of the time from the oscillograph gives $t = 0.795$. The oscillograph for test No. 5A is shown in Fig. 4.

The velocity with which the water strikes the valve at the end of the first resurge is obtained from Equation [8] which gives

$$v_{01} = v' \tanh \frac{g(h + h_0)}{lv'}\, t_2$$

$$= 8.38 \tanh 0.3175 = 2.575 \text{ fps}$$

It will be noted that the value of the exponent n, used in Equation [11], will never change to any extent for a given line, and may be taken as 0.51 for all tests on the 2-in. line; for a long pipe line $n = {}^1/_2$. It is easily shown that if friction and the creation of velocity head are neglected

$$t_1 = \frac{lv_0}{g(h + h_0)}$$

and $t = 2t_1$. In the present instance this would give $t = 0.804$ sec. For low velocities friction can be neglected, the only appreciable loss being that due to shock. In the foregoing, the reduction in velocity due to shock was 0.403 fps, while that due to friction was 0.132 fps.

Second Resurge. Using the same coefficient $C = 0.87$ in the second resurge, we obtain

$$v_0 = 2.575 \times 0.87 = 2.24 \text{ fps}$$

$$t_1 = 1.26 \tan^{-1} (2.24/8.47) = 0.325 \text{ sec}$$

and

$$\cosh 0.802t_2 = [1 + (2.24/8.47)^2]^{0.51} = 1.035$$

from which $0.802t_2 = 0.2642$, and $t_2 = 0.33$ sec. Therefore, $t = t_1 + t_2 = 0.655$ sec. The measurement of the time from the oscillograph gives $t = 0.650$ sec. The velocity at the end of the second resurge is $v_{01} = 8.38 \tanh 0.2642 = 2.165$ fps

Third Resurge. For the third resurge

$$v_0 = 2.165 \times 0.87 = 1.884 \text{ fps}$$

$$t_1 = 1.26 \tan^{-1} (1.884/8.47) = 0.275 \text{ sec}$$

and

$$\cosh 0.802\, t_2 = [1 + (1.884/8.47)^2]^{0.51} = 1.0249$$

from which $0.802t_2 = 0.223$, and $t_2 = 0.278$ sec. Therefore, $t = t_1 + t_2 = 0.553$ sec. The measurement of the time from the oscillograph gives $t = 0.538$ sec. The velocity at the end of the third resurge is $v_{01} = 8.38 \tanh 0.223 = 1.838$ fps.

Fourth Resurge. For the fourth resurge

$$v_0 = 1.838 \times 0.87 = 1.60 \text{ fps}$$

$$t_1 = 1.26 \tan^{-1} (1.60/8.47) = 0.235 \text{ sec}$$

and

$$\cosh 0.802t_2 = [1 + (1.60/8.47)^2]^{0.51} = 1.0181$$

from which $0.802t_2 = 0.1902$, and $t_2 = 0.237$ sec. Therefore, $t = t_1 + t_2 = 0.472$ sec. The measurement of the time from the oscillograph gives $t = 0.467$ sec. The velocity at the end of the fourth resurge is $v_{01} = 8.38 \tanh 0.1902 = 1.575$ fps. The velocities have now become so low that we may neglect friction in computing the next two resurges.

Fifth Resurge. For the fifth resurge

$$v_0 = 1.575 \times 0.87 = 1.37 \text{ fps}$$

and

$$t = 2 \times 1.26 \times (1.37/8.47) = 0.408 \text{ sec}$$

The measurement of the time from the oscillograph gives t = 0.400 sec.

Sixth Resurge. For the sixth resurge

$$v_0 = 1.37 \times 0.87 = 1.192 \text{ fps}$$

and

$$t = 2 \times 1.26 \times (1.192/8.47) = 0.354 \text{ sec.}$$

The measurement of the time from the oscillograph gives t = 0.363 sec.

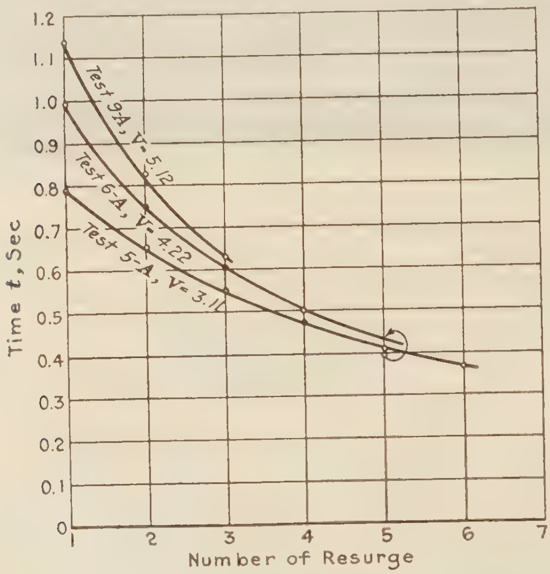

FIG. 5 COMPUTED AND OBSERVED TIME VALUES FOR SUCCESSIVE RESURGES IN TESTS 5A, 6A, AND 9A ON THE 2-IN. LINE

In all the foregoing computations, f has been considered constant, and was selected for the highest value of the velocity. Any error made is easily within the degree of accuracy of the method, particularly as the loss due to f falls off with the velocity.

One difficulty with the method is that the errors are cumulative, each result being based on the one preceding it.

CONCLUSIONS

The results for test No. 5A, just computed, as well as the results for two other tests on the 2-in. line, tests Nos. 6A and 9A, are given in Table 2. The observed results of these three tests, as obtained from the oscillograph are shown graphically by the curves in Fig. 5, while the computed results are shown by the plotted points in Fig. 5.

TABLE 1 DIMENSIONS OF PIPE LINES TESTED

Dimensions	1-in. line	2-in. line
Length l, ft	117.5	278.2
Mean internal diameter d, in.	1.050	2.076
Mean wall thickness t', in.	0.133	0.148

TABLE 2 COMPUTED AND OBSERVED VALUES FOR t, TEST NO. 5A, 6A AND 9A

Resurge	Test No. 5A[a] Computed t	Test No. 5A[a] Observed t	Test No. 6A[b] Computed t	Test No. 6A[b] Observed t	Test No. 9A[c] Computed t	Test No. 9A[c] Observed t
1	0.786	0.795	0.989	1.000	1.135	1.141
2	0.655	0.650	0.762	0.758	0.825	0.816
3	0.553	0.538	0.604	0.602	0.617	0.632
4	0.472	0.467	0.484	0.500	...	...
5	0.408	0.400	0.391	0.425	...	...
6	0.354	0.363	...	...	...	...

[a] For test No. 5A: Line diameter = 2 in., h = 24.1 ft, v = 3.11 fps, C = 0.87, and f = 0.0342.
[b] For test No. 6A: Line diameter = 2 in., h = 24.1 ft, v = 4.22 fps, C = 0.82, f = 0.0312.
[c] For test No. 9A: Line diameter = 2 in., h = 24.1 ft, v = 5.12 fps, C = 0.78, f = 0.031.

TABLE 3 COMPUTED AND OBSERVED VALUES FOR t, TESTS NO. 1 AND 4

Resurge	Test No. 1[a] Computed t	Test No. 1[a] Observed t	Test No. 4[b] Computed t	Test No. 4[b] Observed t
1	0.418	0.420	0.305	0.305
2	0.343	0.342	0.261	0.258
3	0.286	0.283	0.228	0.217
4	0.240	0.225	...	...

[a] For test No. 1: Line diameter = 1 in., h = 24.3 ft, v = 3.73 fps, C for first resurge = 0.93, C for subsequent resurges = 0.88, f = 0.042.
[b] For test No. 4: Line diameter = 1 in., h = 24.3 ft, v = 2.62 fps, C for first resurge = 0.95, C for subsequent resurges = 0.89, f = 0.045.

The relation between the computed and observed values for time can be brought into still closer agreement if, as has been stated previously, a slightly higher value of C be taken for the first resurge, that is, the resurge caused by closing the valve. The difference is more marked in the case of the 1-in. line where, for one initial velocity of v = 3.73 fps, the value of C is 0.93 for the first resurge, and 0.88 for subsequent resurges. In the same line, with v = 2.62 fps, the values of C are 0.95 and 0.89 for the first and subsequent resurges, respectively. Table 3 gives some of the results for the 1-in. line.

ACKNOWLEDGMENT

The author gratefully acknowledges the assistance of M. P. O'Brien, who designed and constructed the laboratory apparatus, and R. G. Folsom, who conducted the experiments on the 2-in. pipe line.

Comparisons Between Calculated and Test Results on Water Hammer in Pumping Plants

By O. SCHNYDER,[1] KLUS, SWITZERLAND

The author discusses surge conditions in pump-discharge lines caused by the sudden failure of the power supply to the pump driving units in both low-head and high-head plants, particularly the latter. He gives a detailed discussion of surge phenomena in pump discharge lines for (*a*) the period immediately following the failure of the power supply when the speed of the pump and discharge both decrease, (*b*) the period during which the reversal of flow continues until maximum discharge is reached in the reverse direction, and (*c*) the period between the maximum backward flow in the pipe line and the complete stoppage of the backward flow. He shows by graphical representation the conditions occurring during period (*b*) when the line is equipped with a flap valve and when it is equipped with a surge suppressor. He concludes the paper with a discussion of surge conditions in the discharge lines of large pumping installations as established by test, and compares such conditions with calculated results, pointing out that the design engineer can predetermine such surge conditions and also calculate accurately the effect of various devices employed to limit water hammer in pumping plants.

THE problem of water hammer in pump-discharge lines has been treated for a long time only by cut-and-try methods. With the development of large motor-driven pumping units, there has been an increasing need for the determination in advance of the surges to be expected under operating conditions.

In a very few cases tests have been carried out in a careful and systematic manner, but in the majority of plants the tests have been limited to making a few measurements after the plant has been completed. In very rare cases research investigations have been made prior to the design and construction of the plant. One of the difficulties in determining the water hammer in pump-discharge lines by calculation has been the general lack of knowledge of pump characteristics. For most pumps no information is available in regard to their behavior under reversed flow or with reversed rotation. It is difficult to compare the performance under test with the theoretical methods of analysis as there is very little reliable information available and practically no laboratory research studies. It is, therefore, necessary to rely entirely upon the meager information obtained from completed plants.

During the last few years, several Swiss pump manufacturers have completed large pumped-storage plants combined with hydroelectric developments. Among these plants are the 4500-hp Sulzer pumps for the Wäggital power plant, and the 13,500-hp high-pressure pumps built by Escher-Wyss for the Tremorgio power plant, operating under a discharge head of about 900 m (approximately 3000 ft). In Germany, the same two companies have constructed plants at Niederwartha, Herdecke, and Schluchsee, in which the pumping units range from 26,000 hp to 40,000 hp. Additional plants, include the Lassoula plant in France, the Urdiceto plant in Spain, and the Zappello plant in Italy. These last three plants are equipped with special quick-closing valves, designed by the Klus Iron Works.

Among the relatively few technical articles published on the subject of water hammer in pump-discharge lines, may be mentioned a contribution of W. Kuhne (1)[2] which describes the operating problems in pumping plants, and also a paper by the author (2).

Where tests were made on the plants previously mentioned the agreement between theory and test was reasonably satisfactory. Where major discrepancies were found, they could be traced to air entrainment in the pipe lines or to the use of unsatisfactory pressure-indicating instruments. In one case, the water contained gas produced by waste materials which reduced considerably the velocity of the pressure waves. In another case, when the power supply to the pump failed, the pressure dropped so rapidly that cavitation conditions were reached within the pump and the discharge pressure dropped abruptly, and the flow also was reduced very rapidly.

In long pipe lines under high pressures, the agreement between test results and theory is much closer than in low-head plants with short pipe lines. With low-head units, the problem of measuring the pressure correctly is of great importance as a time lag or lack of sensitivity in the instruments or the oscillation time of such apparatus may affect the result a great deal and may account for the discrepancies which have been found. The oscillation time of a pressure gage should be at least one tenth that of the critical time ($2L/a$) of the pipe line.

The most important water-hammer problem in motor-driven pumping plants is the surge condition in the discharge lines occurring upon the failure of the power supply to the pump motor. In certain cases, the suction line to the pump is of sufficient length to cause difficulty.

In general, the surge which occurs when starting a pump is usually of less importance than that which occurs during emergency shutdown conditions. The difficulties regarding the starting of pumps can usually be taken care of by use of proper control valves or proper starting equipment, and also by taking particular care to see that all the air has been removed from the discharge lines.

[1] Engineer, Klus Iron Works. Associate Member A.S.M.E. Committee on Water Hammer, representing Switzerland.

Contributed by the Hydraulic Division for presentation at the Second Water-Hammer Symposium, in cooperation with the American Society of Civil Engineers and the American Water Works Association, at the Annual Meeting of The American Society of Mechanical Engineers, to be held at New York, N. Y., December 6-10, 1937.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until March 10, 1938, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] Numbers in parentheses refer to the Bibliography at the end of the paper.

Danger of resonance in the discharge lines is no greater than in turbine penstocks and can usually be avoided by proper construction. By means of the graphical method, an analysis can be made to determine whether or not resonant conditions will occur under normal operation.

As previously mentioned, the principal problem pertains to surges which occur when the power supply to the pump drive suddenly fails. This is divided into two principal cases: (1) Low-head plants with discharge lines having a high velocity of flow, and (2) high-head plants which require close control of the pressure rise.

Low-Head Plants With Discharge Lines Having a High Velocity of Flow

In discharge lines having a high velocity of flow under low heads it is possible, by the use of an ordinary swing-check valve placed at a sufficient distance from the pump, to limit the water hammer to a maximum of twice the normal working pressure without any particular requirements for heavy flywheel effect in the pump motor. A pressure rise in excess of this limit might result if the water column in the pipe line continued to flow up the line after the valve had closed, causing a break in the column with the subsequent heavy return surge. A breaking of the water column in the pump itself should have no serious effect, provided a suitable check valve was installed on the discharge side of the pump. This matter has been discussed in considerable detail by Angus (3).

High-Head Plants Requiring Close Control of Pressure Rise

In high-head plants conditions of economy require that the pressure rise above normal be held to much closer limits than is the case in low-head pumping plants. Careful studies are necessary in order to limit the maximum pressure rise to safe values. The phenomenon which takes place during the surge conditions in pump-discharge lines can be divided into three phases as follows:

First Phase. Immediately upon the failure of the power supply to the pumping unit, the speed of the pump will decrease and the discharge will also decrease. The pressure will drop rapidly and the water in the line will slow down and reverse its direction.

The sudden drop in pressure can cause a break in the water column in the upper part of the pipe line or at any point where the surge gradient would fall materially below the atmospheric pressure. Such cases have been known to rupture the conduit, particularly where the thickness of the pipe in the upper section has been reduced because of the lower operating pressure. To avoid this difficulty it is necessary to arrange the profile of the line to keep a positive pressure at all times and to limit the rate of change of velocity to a safe value.

The pump and motor must have a sufficient flywheel effect of the pipe line or must be provided with an air chamber. It may also be desirable to add check valves, surge tanks, or air-inlet valves at the upper part of the pipe line. The use of air-inlet valves may be a very satisfactory solution in cases where the breaking of the water column cannot be avoided and where other means, such as surge tanks, are too expensive.

It is not possible to set up general rules to cover the solution of the surge problem for all plants and each installation must be studied individually. In many cases, however, the use of a heavy flywheel effect on the pump and motor will be of considerable advantage. In Fig. 1 is shown the relation between the flywheel effect of the pump and driving motor and the fall in pressure in the conduit at the pump. The pressure is shown as the ordinate and expressed as a percentage of the normal working pressure. The abscissas are the relation τ (between the starting time of the pump and the reflection or critical time of the pipe line, $\mu = 2L/a$.

If the starting time of the pump be defined by the formula as $T_a = \theta\omega/M$, where θ represents the flywheel effect or moment of inertia, M the driving moment of the pump and ω the angular velocity. The fall in pressure also depends upon the characteristic of the pipe line, K or ρ determined from the relation aV/gH.

For a given pump at its normal rated capacity, the fall in pressure is determined by both the values τ and K or ρ. This facilitates the comparison between various tests and renders it unnecessary to calculate the fall in pressure for each installation, provided the pumps are similar.

The fall in pressure in the first phase will be a minimum if no restriction takes place in the flow discharged from the pump, but

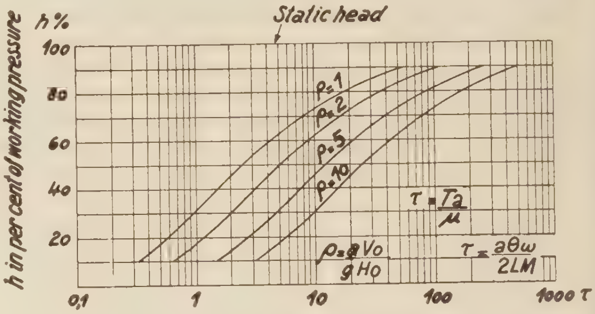

Fig. 1 Minimum Pressure h With Instantaneous Shutting Down of One Pump

if the discharge valve were closed before the water column comes to rest, the down surge would be increased. It is possible to initiate the closing movement of the valve at the instant of the failure of the power supply to the pump, since the first part of the stroke will have little throttling effect upon the discharge, and as the water column slows up the flow through the valve will decrease and the throttling effect will also be reduced.

Surge suppressors, which open automatically upon the failure of the power supply to the pump, may cause a greater fall in pressure than would occur when they are not used.

Second Phase. In the period from the time of reversal of flow until maximum discharge is reached in the reverse direction, there are four cases to be considered, which are discussed in the following paragraphs a, b, c, and d, respectively:

(a) The first case is that with a flap or swing-check valve in the line. In this case a reversal of flow is impossible and the graphical representation is shown in Fig. 2. At the normal point of operation A_0, the pressure drops along the line A_1A_2 until the velocity reaches zero, after which the pressure will surge up and down between A_2 and A_3, and the pressure rise will be equal to the amount the pressure falls below normal. With small flywheel effects, the maximum surge will reach a value of twice the normal head, provided there is no breaking of the water column.

(b) The second case is that with a surge suppressor in combination with a check valve. A different condition would be found if a surge suppressor were used in combination with a check valve and arranged to open as the check valve closes. By carefully regulating the rate of opening of the surge suppressor in relation to the check-valve operation, the pressure rise above normal can be reduced to zero.

Fig. 3 shows the graphical solution of this case, the down surge being the same as in Fig. 2 up to the point A_2 when the surge suppressor opens in accordance with the relation in Fig. 3a. The pressure gradually builds up to normal and the surge suppressor can then be closed slowly enough to hold the pressure rise to some predetermined value.

The maximum velocity in the pipe line due to the flow through the suppressor will be given by the relation $V_s = gh_u/a$, and from this the diameter of the surge suppressor can be calculated.

The same surge suppressor might be operated in accordance with the graphical representation, Fig. 4, in which the opening movement of the surge suppressor is shown in Fig. 4*a*. In this case, the surge-suppressor opening is delayed but finally reaches some position at the end of the time A_3.

It is not always easy to maintain the definite opening rate of the surge suppressor under all conditions and should the valve open too late, an excess surge will be experienced. For this reason it is desirable to initiate the opening movement of the surge suppressor during the first phase, that is, in the time between the ilure of the power supply to the pump and the time with which the velocity in the conduit reaches 0. The valve can continue to open in the second phase, but this will merely increase the area of the orifice. The graphical representation in this case is shown in Fig. 5 and the opening rate of the surge suppressor is indicated in Fig. 5*a*. The suppressor valve starts to open at A_1 and continues up to A_3. The opening rate is limited in the first phase to limit the fall in pressure.

It is advisable to provide the surge suppressor with an adjustable limiting device so that the maximum opening can be

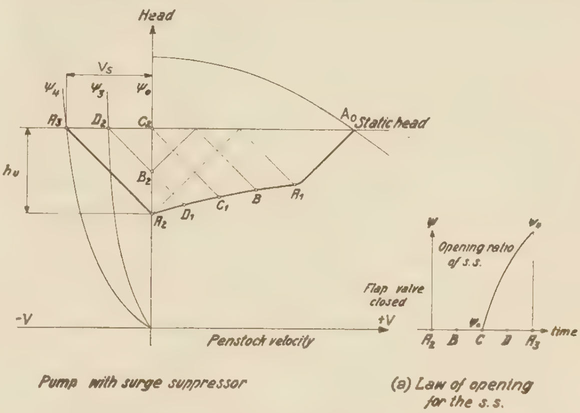

Fig. 4 Pressure Fall in a Pump With a Surge Suppressor the Opening Law of Which Is Represented by the Curve in Fig. 4*a*

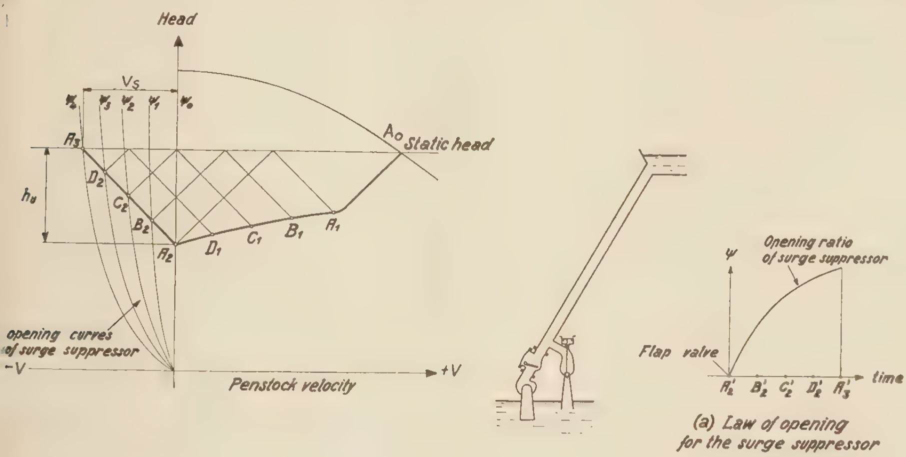

Fig. 3 Pressure Fall in a Pump With a Surge Suppressor the Opening Law of Which Is Represented by the Curve in Fig. 3*a*

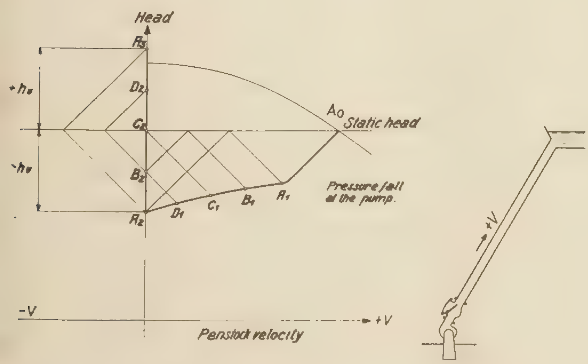

Fig. 2 Pressure Fall in a Pump With a Flap Valve

gulated to suit the particular conditions of the individual inallation.

By using the simple relations outlined, it is possible for the esigner to determine both the flywheel effect and the dimenons of the surge-suppressor valve.

(*c*) Fig. 6 indicates the conditions which exist in the second phase when neither a check valve nor surge suppressor is used and the pump then operates in the reversed direction as a turbine without load. As the speed of the pump decreases and the flow stops, a point is reached where the pump may still be running forward with the flow reversing. The pump will very soon reverse its direction and the general characteristics can be indicated by the family of curves R shown in Fig. 6. The maximum discharge possible will be indicated by the curve P_s, which usually corresponds to the discharge through the impeller with the pump at rest.

Whenever the discharge capacity of the pump during backward flow is greater than would be necessary in a surge-suppressor valve, then a quick-closing check valve can be utilized in the pipe line which can be equipped with a properly selected by-pass that will be kept open during the second phase of the phenomenon. This particular arrangement is shown graphically in Fig. 7.

If pressure still exists on the pump side of the valve, then the area of the by-pass would be increased to compensate for the increased resistance to backward flow. The by-pass area might be fixed approximately so that the discharge through the pump will be equal to or slightly less than the corresponding discharge which would take place through a surge-suppressor valve. In the graphical representation, Fig. 7, this can be determined by adding the frictional resistance of the by-pass to the curve P_s and curves R with the result that another curve P'_s will be obtained.

(*d*) Equivalent action to the combination of the check valve

and by-pass can be obtained with a check valve which can be closed rapidly in the beginning of the stroke, and then more slowly when, toward the end of the first phase, some predetermined opening is reached, the point being fixed at some opening to pass an equivalent discharge to that required in the previous cases (*b*) and (*c*). During the second phase this orifice is maintained open or else closed very slowly until the danger of an excess surge has passed.

Third Phase. This phase is the period between the maximum backward flow in the pipe line and the complete stopping of backward flow.

(*a*) With a quick-closing check valve which prevents return flow, the third phase corresponds with the second phase. The valve is closed and the flow in the line is zero.

(*b*) In cases where a surge-suppressor valve is used, it must be closed very slowly in order to avoid excess pressure rise. The problem involved is similar to the closure of any free-discharge valve and the determination of the closing rate can be made very simply.

(*c* and *d*) In the case where the flow goes back through the pump, the pump itself will have come to rest and reverse in direction, operating as a turbine at runaway speed without load. With the usual type of centrifugal pump, the discharge through the pump will be decreased and a surge above normal will be experienced. The flow at full runaway speed of the pump in a backward direction is represented by the curve P_t in Fig. 8. On the same figure, the curve of discharge versus head for the pump at rest or at maximum backward discharge is shown by the curve P'_s. The further these two curves are apart, the greater the pressure rise will be as a greater reduction in flow takes place during the reversal of the pump. If, by the use of a by-pass or a quick-closing valve held open near the closed position, the condition shown in Fig. 7 will result in a new curve P'_s which is brought nearer to P_t and hence reduces the change in velocity under these conditions. If a sufficiently large flywheel is added, the pressure rise will be reduced since the time required for the velocity change will be lengthened and the surge made less severe.

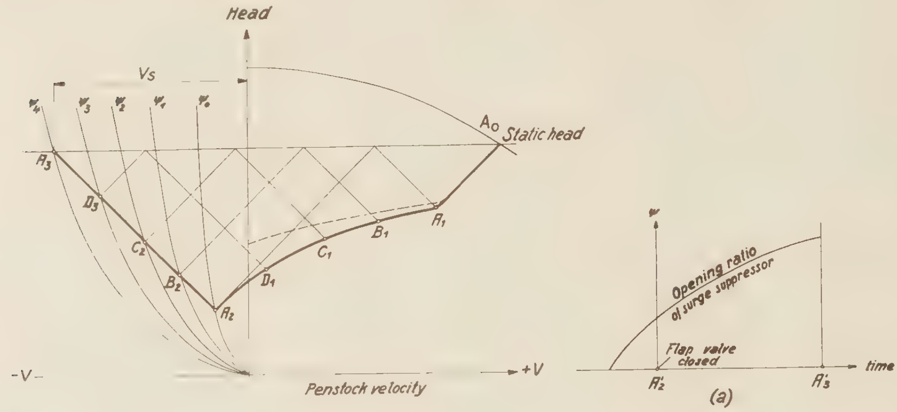

Fig. 5 Pressure Fall in a Pump With a Surge Suppressor the Opening Law of Which Is Represented by the Curve in Fig. 5*a*

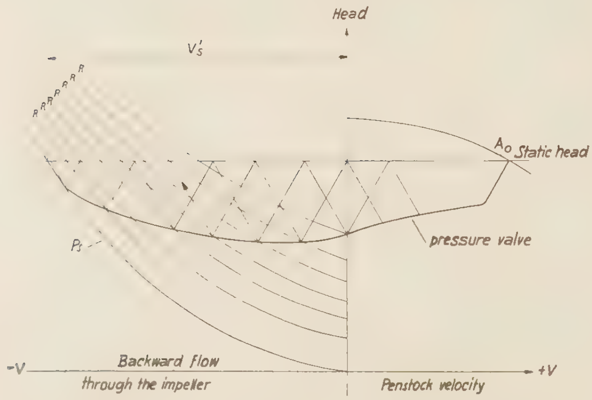

Fig. 6 Pressure Fall in the Second Phase for a Pump Without a Flap Valve

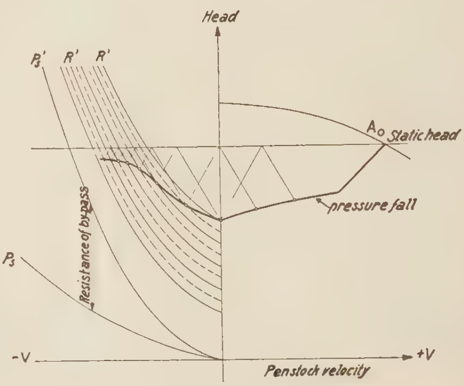

Fig. 7 Pressure Fall in the Second Phase for a Pump With a Flap Valve and a By-Pass

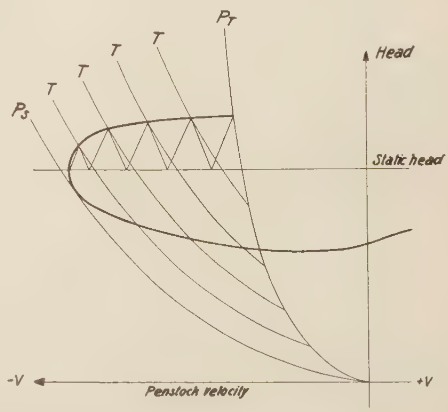

Fig. 8 Pressure Rise in the Third Phase for a Pump Without a Flap Valve

If the pump is equipped with movable guide vanes, the determination of the pressure rise can be made readily by the graphical method, since for each position of the guide apparatus the pump characteristic can be determined, and the closing action of the guide vanes is similar to the operation of a check valve during the third phase.

The preceding explanations will give a reasonably clear idea of the different problems involved in the analysis of water hammer in discharge lines of centrifugal pumps. An understanding of the various features involved will permit a more accurate comparison between the theory and the actual test results. In all of these examples, the effect of pipe line friction or of compound pipes (where variations in thickness and diameter occur) has been neglected since they are a minor factor in this problem.

Comparison of Test Results

In Fig. 9 the variation of pressure with respect to time is shown for a pump discharging into a pipe line without a check valve and where the pump itself introduces the only resistance to backward flow. The flywheel effect of the rotating element is very small, and following the first fall in pressure a very heavy return surge is experienced in the second phase (about 30 per cent above normal).

In Fig. 10 a similar diagram is shown where two pumps are shut down simultaneously in the same discharge line. The fall in pressure and the subsequent pressure rise are not doubled, but the rise is increased to only about 35 per cent above normal as compared with the 30 per cent rise experienced when a single pump is shut down. These correspond very closely to the values obtained by theoretic analyses.

Figs. 9 and 10 also show the conditions in the same plant with a check valve installed to prevent any reverse flow except that which passes through a small by-pass having an area of one eighth that of the pipe line. With the same flywheel effect of the rotating element, the pressure rise during the second phase is reduced materially (about 10 per cent above normal). This is probably due to the reduced rate at which the reversal of the pump rotation is effected, since the by-pass area is small and the effective head on the pump causing reversal is correspondingly reduced.

In certain cases very heavy surges occur because the backward flow is restricted too much in the second phase, or because in the third phase the backward flow is shut off too quickly. In the Wäggital plant, tests have been made on a 5000-hp pumping unit. This plant was originally equipped with a check valve built in the pipe line and fitted with an air brake to retard its movement in the closing direction after the first phase of the down surge. Although an automatic by-pass was in operation, very heavy surges were experienced since the air brake delayed the initial closing of the main check valve and permitted a very heavy backward flow through the pump. When the valve finally closed, considerable velocity was destroyed with a resulting rise in pressure. The results of this test are shown in Fig. 11 as the solid line. A more accurate control was obtained by utilizing a water brake, and the broken line in Fig. 11 indicates the substantial reduction in surge after this change in design. It is estimated that the same result could be obtained if the main check valve was closed completely in the first phase.

Tests which have been made on electrically controlled check valves have given good results and are described in detail elsewhere by the author (2, 4). By the use of surge-suppressor valves the secondary pressure rise can be eliminated almost entirely. A test on equipment of this type has been described by Kerr (5). The fall in pressure following the shut down of the pumps in this plant was increased to some extent by the opening of the surge-suppressor valves during the first phase of the problem.

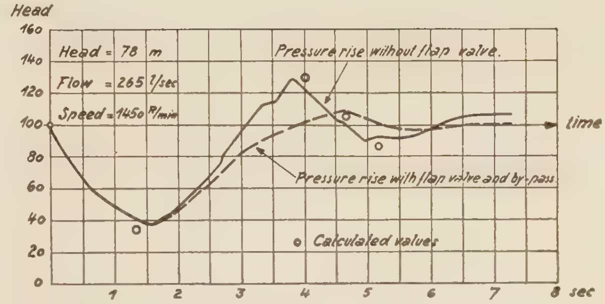

Fig. 9 Water Hammer With Sudden Shutdown of One Pump With and Without a Flap Valve

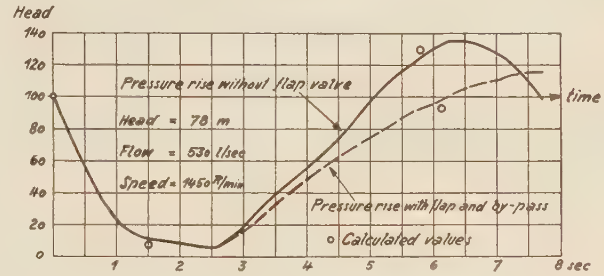

Fig. 10 Water Hammer With Sudden Shutdown of Two Pumps With and Without a Flap Valve

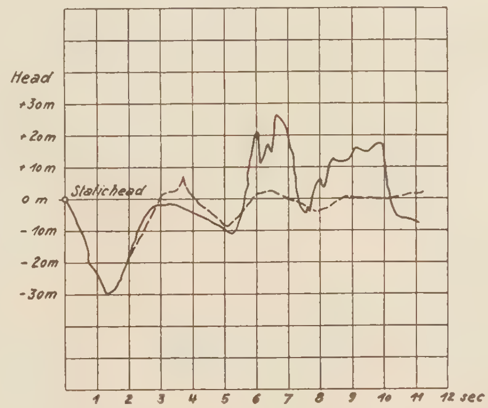

Fig. 11 Water Hammer With Sudden Shutdown of the 5000-Hp Pump in the Wäggital Power Station

(Solid line represents pressure with a check valve equipped with an air brake. Broken line represents pressure rise with the same check valve but equipped with a water brake. Flow = 1118 l per sec; delivery head = 262.8 m; pump power = 5000 hp; and pump speed = 750 rpm.)

A theoretic analysis of the conditions will confirm this statement.

Valuable information has been obtained on several large pumping plants installed by Escher-Wyss & Company in which the pumps are equipped with movable guide vanes for the control of the discharge. The tests on the Niederwartha plant are described by Hermann (6). In this instance the actual test results are compared with the calculations.

The results of the tests on the Schluchsee plant are shown in Fig. 12. Each of the two pressure lines are connected to two pumps and the two discharge lines lead into a common discharge conduit. The test was conducted with one pump shut down instantaneously with the guide vanes partially open. A further analysis was made covering the complete shutdown of all four pumps simultaneously. In each case the arrangement between theory and tests is reasonably satisfactory.

In conclusion, it can be stated that the design engineer now has the means whereby he can predetermine the surge conditions and also calculate accurately the effect of the various devices employed to limit the water hammer in pumping plants. The selection of the proper means to prevent excessive water hammer must be made in each installation with a view to determining the type best suited to the conditions and to provide the maximum protection with a minimum cost.

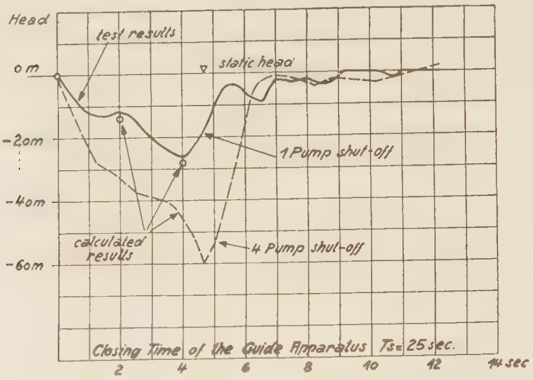

FIG. 12 PRESSURE VARIATIONS IN CASE OF SUDDEN FAILURE OF THE PUMP DRIVE AT THE SCHLUCHSEE PLANT

(Pump characteristics: Delivery head = 200 m; pump speed = 333 rpm; power required for each pump = 23,850 hp; flow = 7.6 cu m per sec; gyration mass GD^2 = 882,000 kg m². Pipe characteristics: Length = 698.4 m; average diameter = 4.25 m.)

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Relation of Relief-Valve and Turbine Characteristics in the Determination of Water Hammer

BY EARL B. STROWGER,[1] BUFFALO, N. Y.

THE GRAPHICAL method of determining the water-hammer pressure in pipes is usable in solving many problems which, with other methods such as arithmetic integration, require considerable time and labor. It has been described in detail within the last few years by engineers of various countries who have written in several different languages. Among the various contributors who have presented many interesting applications of the method to specific problems, the names of Schnyder,[2] Bergeron,[3] and Angus[4] stand out.

In general, the method involves the plotting of sets of simultaneous equations relating flow or velocity to head, and taking the form of a series of straight lines and a series of parabolas, the straight lines representing the pressure due to the direct and indirect water-hammer blows and the parabolas representing discharge through a series of orifices under varying heads.

NOTATION

g = acceleration due to gravity, ft per sec^2
H_0 = pressure head for steady conditions, ft
H = pressure head for surge conditions, ft
h = pressure rise above normal at any time, ft
h' = ratio of total head to initial head, $h' = \dfrac{H}{H_0}$
a = velocity of pressure wave, fps
T = time of gate travel (total), sec
V_0 = velocity in conduit for steady conditions, fps
V = velocity in conduit for surge conditions, fps
V_T = velocity in conduit due to turbine discharge, fps
V_R = velocity in conduit due to relief-valve discharge, fps
v = total penstock velocity ratio at any time, $v = V/V_0$ (max)
v_T = velocity ratio considering turbine discharge, $v_T = V_T/V_0$
v_R = velocity ratio considering relief-valve discharge, $v_R = V_R/V_0$
K = pipe line constant, $K = aV_0/2gH_0$
B_0 = initial gate-opening factor in equation, $V_0 = B_0\sqrt{H_0}$
B_T = gate-opening factor of turbine at any time
B_R = gate-opening factor of relief valve at any time
J = velocity in conduit under normal head (H_0) at gate opening B, $J = B\sqrt{H_0}$
τ_T = ratio of B_T to B_0, $\tau_T = B_T/B_0$
τ_R = ratio of B_R to B_0, $\tau_R = B_R/B_0$
τ = ratio of $(B_T + B_R)$ to B_0, $\tau = (B_T + B_R)/B_0$

Numerical subscripts indicate interval number.

HISTORICAL

As a matter of historical interest, the somewhat parallel development of the graphical method in America and in Europe is mentioned here. The basis of the method was first published by F. M. Wood of McGill University in a technical discussion printed in 1926 and dealing with water hammer in connection with the problem of speed rise of hydraulic turbines.[5] The simultaneous equations used by Wood to develop the method were derived from the water-hammer formulas of Dr. N. R. Gibson. These equations for the first interval were

$$\frac{V_1}{V_0} = \frac{J_1}{V_0}\sqrt{1 + \frac{h_1}{H_0}}$$

$$\frac{h_1}{H_0} = \frac{aV_0}{gH_0}\left(1 - \frac{V_1}{V_0}\right)$$

For the second interval

$$\frac{V_2}{V_0} = \frac{J_2}{V_0}\sqrt{1 + \frac{h_2}{H_0}}$$

$$\frac{h_2}{H_0} = \frac{aV_0}{gH_0}\left(\frac{V_1}{V_0} - \frac{V_2}{V_0}\right) - \frac{h_1}{H_0}$$

And for the nth interval

$$\frac{V_n}{V_0} = \frac{J_n}{V_0}\sqrt{1 + \frac{h_n}{H_0}}$$

$$\frac{h_n}{H_0} = \frac{aV_0}{gH_0}\left(\frac{V_{n-1}}{V_0} - \frac{V_n}{V_0}\right) - \frac{h_{n-1}}{H_0}$$

The Gibson formulas developed in America by Dr. N. R. Gibson in or about the year 1919, were based upon the work of

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[1] Hydraulic Engineer, The Niagara Falls Power Co. Mem. A.S.M.E. Mr. Strowger was graduated in 1918 from the University of Rochester with the degree of B.S. From 1918 to 1919 he was engaged in shipbuilding work for the Interlake Engineering Co. of Cleveland, Ohio. He joined the staff of The Niagara Falls Power Co. in 1919 as a draftsman and designer. At the present time he is engineer of hydraulics with this company. In addition, he has been associated with the chief engineer, Norman R. Gibson in making efficiency tests on hydraulic turbines by the Gibson Method of water measurement.

[2] "Water Hammer in Pump Discharge Lines," by O. Schnyder, *Schweizerische Bauzeitung*, vol. 94, nos. 22 and 23, 1929.

[3] "Variations of Flow in Water Conduits," by L. Bergeron, Comptes Rendues des Travaux de la Société Hydrotechnique de France, 1932, Paris.

[4] "Simple Graphical Solution for Pressure Rise in Pipes and Pump Discharge Lines," by R. W. Angus, *Engineering Journal* (Canada), 1935.

Contributed by the Hydraulic Division for presentation at the Second Water-Hammer Symposium, in cooperation with the American Society of Civil Engineers and the American Water Works Association, at the Annual Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, to be held in New York, N. Y., December 6–10, 1937.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until March 10, 1938, for publication at a later date. Discussion received after the closing date will be returned.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[5] Discussion by F. M. Wood, pp. 243–252, of "Speed Changes of Hydraulic Turbines for Sudden Changes of Load," by E. B. Strowger and S. Logan Kerr, Trans. A.S.M.E., 1926, vol. 48, paper No. 2009, pp. 209–262.

Professor Joukowsky of Moscow, Russia, and were developed quite independently of the work done in Europe by Professor Allievi.

In Europe in 1928, Dr. Loewy of Vienna, Austria, developed the same graphical method independently of the work done by Professor Wood, and this work was described in 1932 in Klus, Switzerland, by Dr. Schnyder[6] who applied the method to the determination of water hammer at intermediate points on the pipe as well as at the gate.

The Governor-Operated Relief Valve

To obtain an accurate determination of the water-hammer pressure which may take place in a penstock, it is necessary to take into account in the computations all the factors which influence the discharge-time curve of the penstock. In the case of a simple penstock installation without branch connections, surge tank, or relief valve, etc., this would involve (*a*) the gate-discharge rating curve of the turbine; (*b*) the gate-time curve of the turbine, and sometimes (*c*) the speed-discharge curve of the turbine although in most cases this last factor may be omitted. To establish these characteristics definitely, it is desirable to make use of acceptance tests or rating tests and sometimes model tests of the turbine.

In the case of a hydroelectric unit with a governor-operated

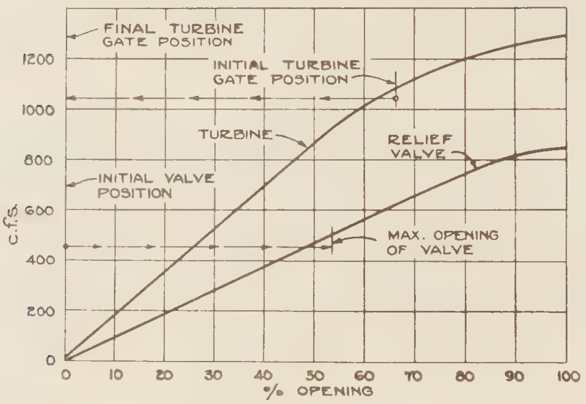

Fig. 1 Relief-Valve and Turbine-Discharge Rating Curves
(Net head 246.4 ft.)

relief valve, the gate-discharge and gate-time curves of the valve must also be known. Furthermore, the action of the turbine must be related to that of the valve for gate movements from various initial positions in order to fix the *maximum* water hammer which may be obtained, because a closure from full gate to zero gate does not necessarily produce the maximum water-hammer pressure, as is shown hereinafter.

Using the suffix T to refer to the turbine and R to refer to the relief valve, then V_T defines the velocity in the penstock caused by the turbine discharge and V_R the velocity in the penstock caused by the relief valve. We may use lower-case letters for indicating velocity, head, etc., in terms of relative values. These relative values relate quantities at any instant to the initial quantities. Thus

$$v_T = \frac{V_T}{V_0},\ \tau_T = \frac{B_T}{B_0} \text{ and } h' = \frac{H_T}{H_0}$$

Then for the turbine

$$v_T = \frac{V_T}{V_0} = \frac{B_T}{B_0}\sqrt{\frac{H}{H_0}} = \tau_T\sqrt{h'} \ldots\ldots\ldots\ldots [1]$$

For the relief valve B_R may be related to the initial turbine-gate opening factor B_0 and H_R to H_0 as follows:

$$v_R = \frac{V_R}{V_0} = \frac{B_R}{B_0}\sqrt{\frac{H}{H_0}} = \tau_R\sqrt{h'} \ldots\ldots\ldots [2]$$

For the penstock

$$v = v_T + v_R = (\tau_T + \tau_R)\sqrt{h'} \ldots\ldots\ldots\ldots [3]$$

Computation Neglecting Friction

A computation for a 54-in. governor-operated relief valve installed on a 29,000-hp hydraulic turbine is given below. No surge tank was installed in connection with the turbine. The pertinent physical data of the plant are as follows:

Length of penstock, ft	3969
Velocity of pressure wave, fps	2300
Initial gross head, ft	260.9
Initial net head, ft	246.4
Initial discharge, cfs	1075
Initial load, kw	20,000

$$\frac{2L}{a} = 3.45 \text{ sec}$$

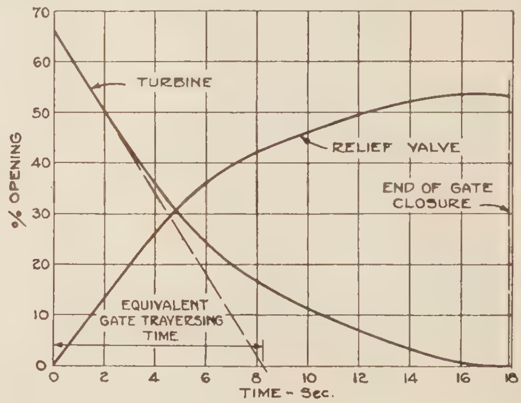

Fig. 2 Gate-Time Curves
(Net head 246.4 ft; 66 per cent gate to zero.)

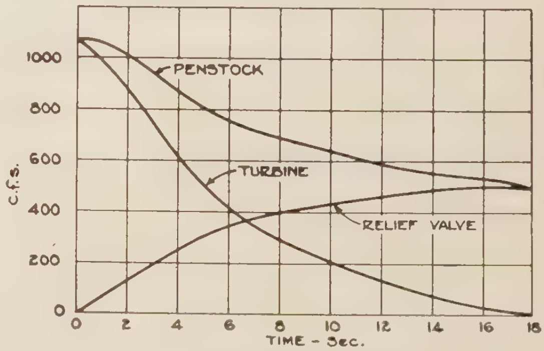

Fig. 3 Discharge-Time Curves
(Net head, 246 ft; 66 per cent to zero.)

The gate-discharge curves of the turbine and relief valve are shown in Fig. 1 and one of the gate-time curves in Fig. 2. The resulting discharge-time relations for a uniform net head of 246.4 ft are shown in Fig. 3 which also gives the penstock discharge-

[6] "Uber Druckstosse in Rohrleitungen," by O. Schnyder, *Wasserkraft und Wasserwirtschaft*, March, 1932.

TABLE 1 GATE MOVEMENT 66 PER CENT TO ZERO

Time: Intervals	Time: Seconds	Discharge cfs $h' = 1$: Turbine	Discharge cfs $h' = 1$: Relief valve	Discharge cfs $h' = 1$: Penstock	τ	Values of v: $h' = 1.137$	Values of v: $h' = 1.298$	Values of v: $h' = 1.461$
0.0	0.0	1075	0	1075	1.0	1.066	1.138	1.208
0.25	0.86	1010	51	1061	0.987	1.053	1.125	1.193
0.50	1.73	910	110	1020	0.949	1.012	1.080	1.147
0.75	2.59	798	162	960	0.893	0.952	1.016	1.079
1.00	3.45	685	220	905	0.842	0.897	0.958	1.018
1.25	4.31	575	267	842	0.783	0.834	0.892	0.946
1.50	5.18	489	310	799	0.743	0.792	0.846	0.898
1.75	6.04	413	344	757	0.704	0.750	0.802	0.850
2.00	6.90	349	370	719	0.669	0.713	0.762	0.808
2.25	7.76	304	390	694	0.645	0.687	0.734	0.779

time curve assuming a uniform head. From this curve, Table 1 has been prepared by using Eq [3] to obtain the family of parabolas giving discharge and head from the particular gate opening obtained at each interval point.

These parabolas have been plotted in Fig. 4 which is the

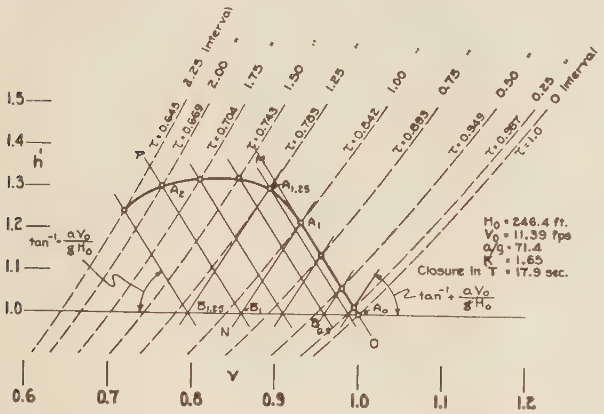

FIG. 4 v-h-τ DIAGRAM FOR TURBINE WITH GOVERNOR-OPERATED RELIEF VALVE

(Closure 66 per cent gate to zero. Friction neglected.)

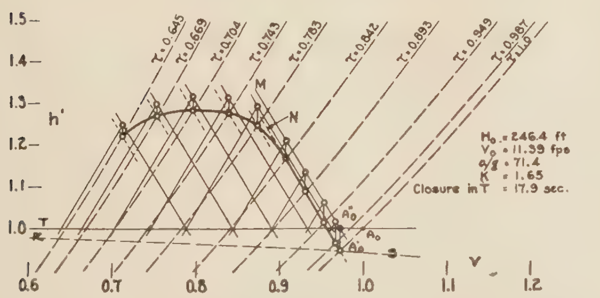

FIG. 5 v-h-τ DIAGRAM FOR TURBINE WITH GOVERNOR-OPERATED RELIEF VALVE

(Closure 66 per cent to zero. Friction considered.)

v-h-τ diagram neglecting friction. The lines OM, B_1P, etc., represent the direct water-hammer blows and have a slope equal to $-aV_0/gH_0$ as may be shown as follows. The relation between pressure rise in feet of water and velocity change fps is

$$\Delta H = -\frac{a}{g} \Delta V$$

Whence

$$\frac{H_0 + \Delta H}{H_0} = \frac{H}{H_0} = h' = -\frac{a\,\Delta V}{gH_0} + 1$$

and

$$h' - 1 = -\frac{a\,\Delta V}{gH_0}$$

But the slope of a straight line drawn to coordinates of h and v is equal to

$$\frac{h' - 1}{\Delta v} = \frac{h' - 1}{\Delta V / V_0}$$

where Δv is measured from the initial point (1.0, 0) and therefore this slope must be

$$-\frac{a\,\Delta V}{g\,H_0}\,\frac{V_0}{\Delta V} = -\frac{aV_0}{gH_0}$$

The lines B_1A_1, $B_{1.25}A_{1.25}$, etc., represent the indirect water-hammer blows and have a slope equal to

$$+\frac{aV_0}{gH_0}$$

The A points represent the conditions at the turbine and the B points represent the conditions at the forebay of the power plant.

After the parabolas have been drawn, the line OM is drawn with a slope equal to $-\frac{aV_0}{gH_0}$ and passing through the point A_0 which has the coordinates (1.0, 0). The intersections of the line OM with the parabolas $\tau = 1.0$, $\tau = 0.987$, $\tau = 0.949$, $\tau = 0.893$,

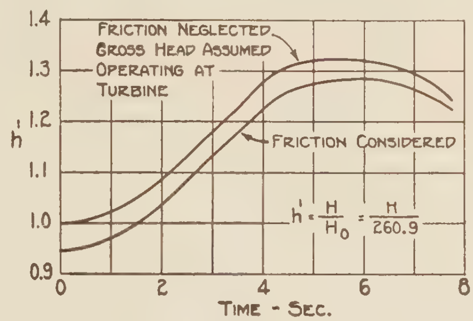

FIG. 6 PRESSURE-TIME CURVES AT THE TURBINE AND RELIEF VALVE FOR FIGS. 4 AND 5

and $\tau = 0.842$ give the head and velocity conditions at intervals 0, 0.25, 0.50, 0.75, and 1.0, respectively.

The A points in the second interval are obtained by drawing lines through corresponding points of the first interval having a slope of $+\frac{aV_0}{gH_0} = +3.30$ and determining the points of intersection with the line $h' = 1$ from which points new slope lines parallel to OM are drawn intersecting the parabolas representing the corresponding phase in the second interval, i.e., the point A_2 is found by drawing A_1N through A_1 and with slope $+3.30$ determining B_1 through which the line B_1P is drawn intersecting the parabola for the second interval at A_2.

COMPUTATION CONSIDERING FRICTION

Fig. 5 is presented to show one method of making the diagram with friction taken into account. The method has been proposed by Professor Bergeron[3] and is approximate in that it assumes all the friction concentrated at one point in the line. The diagram is similar to the one given by Fig. 4 except that the line RS is drawn below the line $h' = 1$ by a distance representing the friction loss in the pipe line and instead of using one set of working points a second set placed below the first set by a distance equal to the friction loss gives the true pressure rise. Referring to Fig. 5 the initial point A_0 as given in Fig. 4 is placed at A'_0 which is located at the intersection of the line RS and the parabola $\tau = 1.0$. From a point A''_0 on the line $h' = 1$ and

vertically above A'_0, the slope line A''_0M is drawn. The curved line NA'_0 is then drawn below A''_0M by the amount of the friction loss as measured by the vertical distance between RS and TA_0. The intersections of the line NA'_0 with the various parabolas give the head and velocity conditions at the intervals represented by the τ lines. The points immediately above these points and located on the straight line MA''_0 are the working points to use in drawing the lines of positive slope representing the indirect water-hammer blows and extending down to the line TA_0 to determine points from which to draw more lines of negative slope to determine the head and velocity conditions during the second interval, etc.

If it is desired to take into account the velocity head at the entrance to the turbine, it may be combined with the friction loss in the pipe line in determining the line RS which in this case

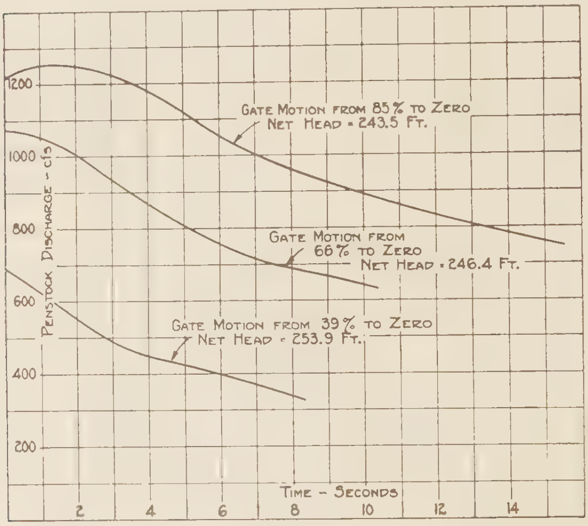

FIG. 7 DISCHARGE-TIME CURVES OF TURBINE AT START OF GATE CLOSURE

would represent the recovery of friction and velocity heads as closure takes place. In either case, at $v = 0$, the line RS will pass through $h' = 1.0$.

The v-h-τ diagram neglecting friction indicates a pressure rise of 32 per cent, and the one taking friction into account gives a pressure rise of about 28 per cent.

Fig. 6 shows the pressure-time curve at the turbine and relief valve for Figs. 4 and 5.

EFFECT OF SHAPE OF GATE-DISCHARGE CURVE AND OF GATE MOTION

Fig. 1 shows that in the range of high turbine gate opening the gate-cfs curve is rather flat, which would result in a small initial rate of decrease of v_T with closure starting from such a gate opening. On the other hand, the rate of increase in v_R is a maximum at the start of the motion. The combination of these two factors may result in creating an increase in v at the start of the motion. This tendency is indicated by the penstock discharge curve of Fig. 3 which shows a small increase in v at the start. Fig. 7 shows the shape of the discharge-time curves of the turbine at or near the start of closure for three different gate openings. The curve for a gate movement of 85 per cent to zero shows a considerable increase in penstock velocity at the beginning of the movement. The effect of this particular gate motion is shown by the v-h-τ diagram of Fig. 8 which indicates that a negative wave first starts up the pipe.

FIELD TESTS

Pressure-rise tests made on the 29,000-hp turbine and 54-inch relief valve before mentioned were made in 1936. The results of these tests may prove to be interesting in connection with the computations of pressure rise made to check them. They consisted in measuring the maximum pressure rise at the turbine and the speed rise of the unit accompanying the tripping of various loads off the unit. Electric boilers used for loading the unit allowed the selection of any load desired within the capacity of the

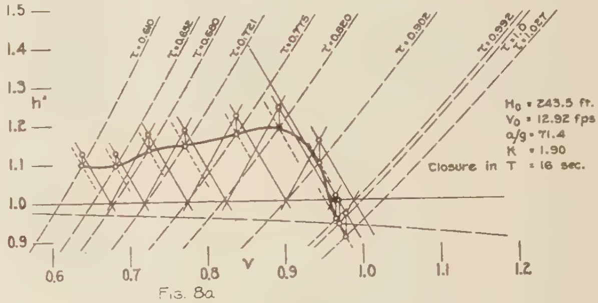

FIG. 8(a) v-h-τ DIAGRAM FOR TURBINE GOVERNOR-OPERATED RELIEF VALVE

(Closure 85 per cent gate to zero.)

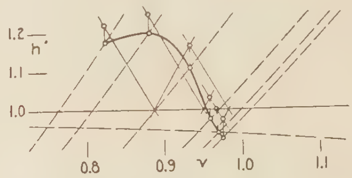

FIG. 8(b) FIG. 8(a) CORRECTED FOR SPEED-DISCHARGE CHARACTERISTIC OF TURBINE

unit. The loads actually tripped varied from 8000 to 21,900 kw. The pressure rise was measured by each of three calibrated gages, one of which was a special test gage, and the speed rise was measured by a calibrated tachometer. The rate of gate motion was measured by means of an instrument called a "gate-opening recorder" which comprised a drum driven by a motor of constant speed, a sheet of "metallic recording paper" mounted on the drum, and a metallic pointer attached to the servo-motor mechanism and arranged to mark on the paper. This device accurately determined the position of the gate at any instant during the closing motion, the position just prior to the shutdown, as well as the time of duration of the gate stroke.

The value of a had been previously determined by tests made with a pressure gage and a small motion-picture camera by obtaining one complete interval of a square-topped wave. The wave was produced by closing the turbine from 10 or 11 per cent gate to zero in six or seven seconds. The value of a for the whole pipe was determined to be about 2400 fps. By computation the values of a for the intake, the wood-stave pipe, and the steel pipe were determined to be as follows:

	Length, ft	a, fps
Intake	46	4600
Wood-stave	2990	2040
Steel	932	2900

The computed value of a for the combined pipe is 2200 fps. For the purpose of making the water-hammer computations the value of a selected for the whole pipe has been taken at 2300 fps.

The relatively high value of a for the wood-stave line is accounted for by the heavy construction of the pipe. The staves

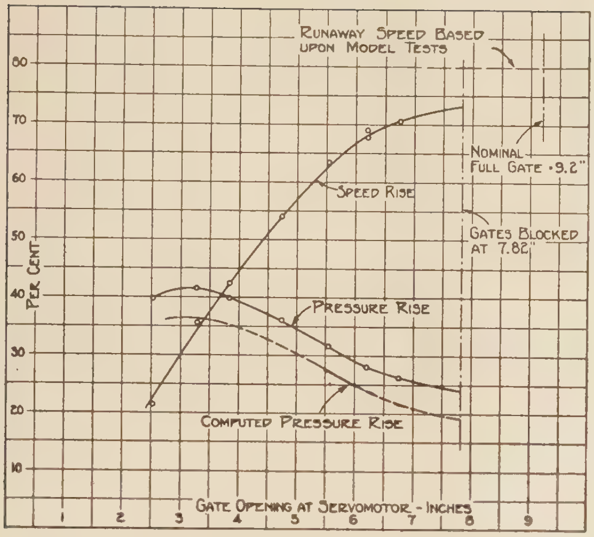

Fig. 9 Comparison of Test and Computed Results

varied from $3^1/_2$ in. thick at the upper end to 5 in. thick at the lower end and the area of bands per ft of pipe varied from 0.885 sq in. to about 5.96 sq in.

Fig. 9 shows the results of the regulation and pressure-rise tests, and also gives the results of the computation on water hammer. The curve shows the maximum pressure rise for each of the various initial gate openings from which the gate has traveled to the zero position. Each point on the curve represents a closure from some particular gate opening and gives the maximum points which would be taken from curves of pressure with respect to time or speed with respect to time. The agreement, in general, is within about 4 to 5 per cent of the pressure rise and the shape of the computed curve is very nearly the same as the experimental one. Some of the difference may be accounted for by taking into account the influence of the speed change on the runner discharge as discussed later. A maximum pressure rise of 42 per cent was obtained with a closure from about 3.3 in. servo-motor stroke or about 36 per cent gate opening, at which point the equivalent gate traversing time of the governor is about 5 sec or about $1^1/_2$ intervals.

The speed-rise curve suggests that the turbine discharge-speed characteristic should perhaps be taken into account in computing the water hammer. The increase in speed of the turbine may cause the parabolas of the v-h-τ diagram to change their positions slightly. The effect of the speed rise in the case of the closure represented by Fig. 8(a) was to cause the parabolas to move slightly to the left. The net result was an increase of about 1 per cent in the water-hammer pressure over that already computed.

The Application of Heaviside's Operational Calculus to the Solution of Problems In Water Hammer

By F. M. WOOD,[1] MONTREAL, CANADA

The author points out that Heaviside's operational calculus, which has been used extensively in connection with electrical surges in transmission lines, may be used in the study of water hammer even though certain limitations are involved because of necessary approximations which must be made when applying it to hydraulics. In spite of these limitations, the method is useful in clarifying many surge problems, verifying some truths which have been established by other methods. It is not the purpose of this paper to attempt to cover the entire field of surge problems, but to introduce Heaviside's operational calculus in the field of hydraulics for the study of water hammer.

THE nomenclature used in this paper is as follows:

D = diameter of pipe
L = length of pipe
A = area of pipe
P = pressure, lb per unit area
V = velocity
x = distance of section along pipe
K = volume modulus of compression of water
b = thickness of pipe walls
E = modulus of elasticity of pipe walls
f = friction coefficient
w = weight of unit volume of water
g = acceleration due to gravity
k_f = friction factor for linear approximation
Z = static head
$W = w/g;\ Q = (1/K) + (D/bE)$
p = operator d/dt
t = time variable
T = particular time
a = velocity of pressure wave
μ = period of pipe line $= 2L/a$

Heaviside's operational calculus has been used extensively in connection with electrical surges in transmission lines, and since the subject of water hammer is an analogous one, it may prove amenable to similar applications of the method. However, there are certain limitations of the method which should be mentioned at the outset, but which should not prevent the attempt from being made since all theories suffer from similar handicaps, in one way or another. In the practical solution of particular cases, so many conditions of an indeterminate nature are present that approximations are inevitable. Any theory to be of value should take into account the important facts of the problem, and make use of approximations to a minimum extent. The more true to facts and the fewer the approximations, the better the theory. In hydraulic problems, approximations are found necessary for two main reasons: (*a*) the inability of the mathematical theory to solve the complete equations of flow (e.g., friction terms), and (*b*) peculiar characteristics of the various hydraulic machines involved (e.g., pump and motor characteristics, or governor action in a turbine). The use of graphical methods in solving the theoretical equations for successive short intervals of time has been found to be of great help.

The Heaviside calculus was developed for electrical problems, and can solve only linear differential equations (linear networks in electrical problems). In hydraulics it is usual to assume friction to vary as the square of the velocity, making the friction term in our equations a quadratic term; hence, an approximation is necessary here. Again, the Heaviside method requires the sudden application at zero time of a constant pressure (or velocity), or of a pressure (or velocity) which can be expressed as a known function of time. In problems where this is not the case (e.g., problems involving the characteristics of a pump or turbine) approximations are necessary.

In spite of these limitations, this method is useful in clarifying many surge problems—verifying some truths which have already been established by other methods. It is not the purpose of this article to attempt to cover the entire field of surge problems, but to introduce this valuable tool to the hydraulic fraternity. In the various examples, no detailed study will be given of the operational part of the work; however, the bibliography given at the end of the paper should be of help to those desiring to investigate this method further.

The Fundamental Differential Equations of Flow in Pressure Channels. In Fig. 1 are shown the conditions of flow at two successive instants T_1 and T_2; the element $AB = \Delta x$ has moved to CD.

Denoting the pressure and velocity at A by P and V, respectively, then

$$P_C = P + \frac{\partial P}{\partial t}\,\Delta t + \frac{\partial P}{\partial x}\,A_1C_2;\ P_B = P + \frac{\partial P}{\partial x}\,\Delta x$$

[1] Assistant Professor of Civil Engineering, McGill University. Associate Member, A.S.M.E. Committee on Water Hammer, representing Canada. Professor Wood was graduated from Queens University, Kingston, Ontario, Can., in 1911 with an M.A. degree, and received a B.Sc. degree from the same university in 1914. He served with the Canadian Field Artillery and the Royal Air Force in France during the World War. From 1919 to 1921 he was a lecturer at Queens University, and from 1921 to 1923 was engaged in irrigation construction work in the Lethbridge Northern Irrigation District, Alberta, Can. He served in the hydraulic department of the Dominion Engineering Works, Montreal, Canada, from 1923 to 1925, and since that date has been at McGill University.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

$$P_D = P + \frac{\partial P}{\partial x}\Delta x + \frac{\partial P}{\partial t}\Delta t + \frac{\partial P}{\partial x} B_1 D$$

Similar expressions will give the various velocities.

Since the diameter of the pipe varies little in comparison with the pressure (of the order of 1/2000 in ordinary pipe lines) it may safely be assumed that the diameter or area of the pipe is constant in comparison with pressure.

It may also be shown that the reduction in length of an element of water Δx due to an increase in pressure ΔP is $\Delta x[(1/K) + (D/bE)]\Delta P$. Also

$$A_1C = V\Delta t\left[1 + \left(\frac{1}{2V}\frac{\partial V}{\partial t} + \frac{1}{2}\frac{\partial V}{\partial x}\right)\Delta t\right]$$

$$B_1D = \left(V + \frac{\partial V}{\partial x}\Delta x\right)\Delta t\left[1 + \left(\frac{1}{2V}\frac{\partial V}{\partial t} + \frac{1}{2}\frac{\partial V}{\partial x}\right)\Delta t\right]$$

Here all terms have been retained and are canceled where permissible at the end of the analysis.

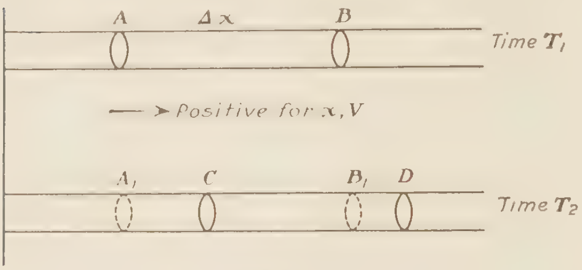

Fig. 1

There are two conditions of flow: (a) The resultant force acting on the element is equal to its time rate of change of momentum, and (b) the element AB must compress into the element CD.

Assuming a uniform horizontal pipe, and substituting our various expressions for the pressures and velocities, assuming also that the force of friction is $(4fL/D)(V^2/2g)$ feet of water, or $(4fwA/2Dg)V^2\Delta x$ lb, the two conditions of flow reduce to

$$-\frac{\partial P}{\partial x} - \frac{2fw}{gD}V^2 = \frac{w}{g}\left[\frac{\partial V}{\partial t} + V\frac{\partial V}{\partial x}\right] \ldots\ldots\ldots [1]$$

$$-\frac{\partial V}{\partial x} = \left(\frac{1}{K} + \frac{D}{bE}\right)\left[\frac{\partial P}{\partial t} + V\frac{\partial P}{\partial x}\right] \ldots\ldots\ldots [2]$$

The terms $V(\partial V/\partial x)$ and $V(\partial P/\partial x)$ should be noted. Usually the partial derivatives with respect to x are small compared to the corresponding partial derivatives with respect to time, so that these terms may with safety be neglected. Solutions for uniform pipe lines have been worked out by the author with these terms included, but wherever they are found they involve the extra factor $[(1/K) + (D/bE)]$ which is known to be negligible compared to unity.

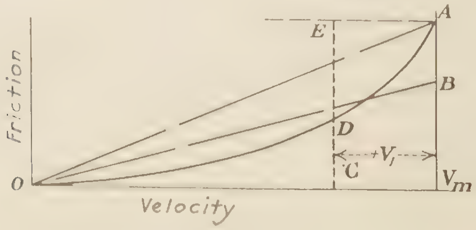

Fig. 2

The nonlinear terms are $[(2fw/gD)V^2]$, $V(\partial V/\partial x)$, and $V(\partial P/\partial x)$. The friction term is the only important one, and requires an approximation. In these studies, certain examples will be solved first neglecting friction, and later assuming friction to be linear or varying directly with velocity.

For this purpose

$$\frac{2fw}{gD}V^2 = \frac{2fw}{gD}(k_f V_m)V$$

For $k_f = 1$, the friction is represented by OA in Fig. 2; for $k_f = {}^2/_3$, it is represented by OB. The latter assumption (OB) would appear to be most accurate where the velocities ranged from O to V_m, and the former (OA) where only zero velocity and V_m are involved. One thing in favor of the use of a linear friction term is that for reversal of flow the sign of the friction term changes automatically. Development of theoretical solutions with a quadratic friction term are difficult in cases of reversal of flow.

Equations [1] and [2] are true for a horizontal pipe line. For an inclined pipe, the pressure P must be replaced by the sum of the pressure and static head, i.e., for P write $P + Zw$. The various examples constitute the remaining portion of this paper.

Example 1

Use the simple uniform horizontal pipe of length L, diameter D, running full under pressure from a reservoir with velocity V_m. Let the gate suddenly be closed at the lower end of the pipe. Friction is to be neglected in this example.

By applying a sudden constant velocity, $-V_m$ at time zero, at the lower end, the solutions will give the resulting surges of pressure and velocity, which must be superimposed on the constant steady-flow conditions to obtain the actual pressures and velocities during the surging.

The simplified equations of flow are

$$-\frac{\partial P}{\partial x} = \frac{w}{g}\frac{\partial V}{\partial t} = W\frac{\partial V}{\partial t} \ldots\ldots\ldots\ldots\ldots [3]$$

$$-\frac{\partial V}{\partial x} = \left(\frac{1}{K} + \frac{D}{bE}\right)\frac{\partial P}{\partial t} = Q\frac{\partial P}{\partial t} \ldots\ldots\ldots\ldots [4]$$

Using P for pressures, and p for the operator (d/dt), these lead to

$$\frac{\partial^2 P}{\partial x^2} = -W\frac{\partial^2 V}{\partial x\,\partial t} = W\frac{\partial}{\partial t}\left(Q\frac{\partial P}{\partial t}\right) = WpQpP = WQp^2P \ldots [5]$$

The operational solution of this is

$$P = A_1 \cosh px\sqrt{WQ} + B_1 \sinh px\sqrt{QW} \ldots\ldots\ldots [6]$$

where A_1 and B_1 are arbitrary constants to be determined by the conditions of surge. For velocity there is $W(\partial V/\partial t) = -(\partial P/\partial x)$ or

$$WpV = -p\sqrt{WQ}\left(A_1 \sinh px\sqrt{WQ} + B_1 \cosh px\sqrt{WQ}\right) . [7]$$

which reduces to

$$V = -\sqrt{Q/W}\left(A_1 \sinh px\sqrt{WQ} + B_1 \cosh px\sqrt{WQ}\right) . . [8]$$

Insert now the surge conditions $P = 0$ at intake $x = 0$ (i.e., no pressure surge at intake), and $V = -V_m$ at gate $x = L$ (i.e., this suddenly applied velocity at lower end results in actual zero velocity during the surge). These two conditions give $A_1 = 0$ and $V_m = \sqrt{(Q/W)}B_1 \cosh pL\sqrt{WQ}$. Substituting for A_1 and B_1 in Equations [6] and [8] gives the operational solutions

$$P = \frac{V_m\sqrt{(W/Q)} \sinh px\sqrt{WQ}}{\cosh pL\sqrt{WQ}} \ldots\ldots\ldots\ldots [9]$$

$$V = -\frac{V_m \cosh px \sqrt{WQ}}{\cosh pL \sqrt{WQ}} \qquad [10]$$

The interpretation of the operator $\left[(\sinh px\sqrt{WQ})/(\cosh pL\sqrt{WQ})\right]$ is shown in Fig. 3. Here the time unit $T = 2L\sqrt{WQ}$ and the breaks in the function occur at intervals $\pm[(L-x)/(2L)]T$ on either side of the instants T, $2T$, $3T$, etc. At the section x from the intake, there is no surge until the time $t = [(L-x)/(2L)]T$, the pressure then surges up an amount $V_m\sqrt{W/Q}$, and after that alternates down and up as shown. A complete cycle occurs in time $2T$, and the complete picture is that of a positive pressure wave traveling up the line at a velocity of $(L-x)$ ft in $[(L-x)/(2L)]T$ sec, or $2L/T = 1/\sqrt{WQ}$ fps. When this wave reaches the intake it sets up a return negative wave of the same magnitude which cancels the first wave as it proceeds downstream. Then a negative wave starts up the line followed by a return positive canceling wave. The agreement with Allievi's solution will be recognized and it is possible in the future to denote the velocity of the pressure wave $1/\sqrt{WQ}$ by a. The magnitude of the pressure rise is $V_m(W/\sqrt{WQ}) = V_m(w/g)a$ lb or aV_m/g ft of head, as expected.

The interpretation of the operator $[(\cosh px\sqrt{WQ})/(\cosh pL/\sqrt{WQ})]$ is shown in Fig. 4. The breaks occur here at intervals of $\pm[(L-x)/(2L)]T$ from times T, $2T$, $3T$, etc. as for pressure surges. If we superpose these velocity surges of $-V_m$ and $-2V_m$ on the steady initial velocity $+V_m$, the resulting velocity during the surge alternates between $+V_m$ and $-V_m$. Before leaving this example, it may be useful to explain the interpretation of these operators, and their connection with the traveling waves. The pressure operator ($\sinh \alpha x/\cosh \alpha L$), where $\alpha = p\sqrt{WQ}$, may be expanded to

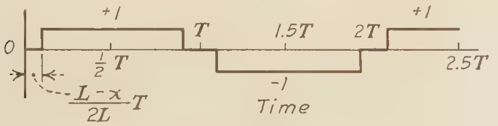

Fig. 3

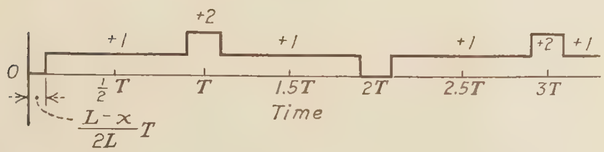

Fig. 4

$$\frac{e^{\alpha x} - e^{-\alpha x}}{e^{\alpha L} + e^{-\alpha L}} = e^{-\alpha(L-x)} \frac{1 - e^{-2\alpha x}}{1 + e^{-2\alpha L}}$$
$$= [e^{-\alpha(L-x)} - e^{-\alpha(L+x)}][1 + e^{-2\alpha L}]^{-1}$$
$$= [e^{-\alpha(L-x)} - e^{-\alpha(L+x)}][1 - e^{-2\alpha L} + e^{-4\alpha L} - e^{-6\alpha L} + \ldots]$$

The operator $e^{-\alpha(L-x)} = e^{-p(L-x)\sqrt{WQ}}$ signifies a delay in time of $(L-x)\sqrt{WQ}$; up to that time it is equal to zero, after that time equal to unity. Hence, the first bracket represents the positive wave, inoperative until the time $(L-x)\sqrt{WQ}$, and the canceling return wave inoperative until the time $(L+x)\sqrt{WQ}$. Each successive term in the second bracket represents successive double waves (direct and return).

Example 2

In this example, consider the compound pipe shown in Fig. 5 with the gate at the lower end suddenly closed. The upper and lower sections of the pipe L are denoted by the subscripts 1 and 2, respectively. The general operational equations, neglecting friction are

$$P_1 = A_1 \cosh px\sqrt{WQ_1} + B_1 \sinh px\sqrt{WQ_1} \qquad [11]$$

$$P_2 = A_2 \cosh px\sqrt{WQ_2} + B_2 \sinh px\sqrt{WQ_2} \qquad [12]$$

$$V_1 = -\sqrt{Q_1/W}\,(A_1 \sinh px\sqrt{WQ_1} + B_1 \cosh px\sqrt{WQ_1}) \qquad [13]$$

$$V_2 = -\sqrt{Q_2/W}\,(A_2 \sinh px\sqrt{WQ_2} + B_2 \cosh px\sqrt{WQ_2}) \qquad [14]$$

The conditions of flow are: $P_1 = 0$ at $x = L_1$; $V_2 = -V_m$ at $x = L_2$ (assuming velocity initially at lower end is V_m); $P_1 = P_2$ at $x = 0$; and $D_1^2V_1 = D_2^2V_2$ at $x = 0$. Note that these are the continuous conditions of flow after the gate has been closed.

Substituting these conditions and solving for the arbitrary constants A_1, A_2, B_1, and B_2, the following pressure solutions are obtained

$$P_1 = \frac{V_m\sqrt{W/Q_2}\,\sinh \alpha_1(L_1 + x)}{\sinh \alpha_1 L_1 \sinh \alpha_2 L_2 + (D_1^2/D_2^2)\sqrt{(Q_1/Q_2)}\cosh \alpha_1 L_1 \cosh \alpha_2 L_2} \qquad [15]$$

$$P_2 = \frac{V_m\sqrt{W/Q_2}\,[\sinh \alpha_1 L_1 \cosh \alpha_2 x + (D_1^2/D_2^2)\sqrt{Q_1/Q_2}\cosh \alpha_1 L_1 \sinh \alpha_2 x]}{\sinh \alpha_1 L_1 \sinh \alpha_2 L_2 + (D_1^2/D_2^2)\sqrt{Q_1/Q_2}\cosh \alpha_1 L_1 \cosh \alpha_2 L_2} \qquad [16]$$

Equations [15] and [16] must be expanded into series of exponentials. Denoting by C the expression

$$\frac{D_1^2\sqrt{Q_1} - D_2^2\sqrt{Q_2}}{D_1^2\sqrt{Q_1} + D_2^2\sqrt{Q_2}}$$

then the expansions for the first few terms are

$$P_1 = V_m\sqrt{W/Q_2}\,(1-C)[e^{-\alpha_2L_2 + \alpha_1x} - e^{-\alpha_2L_2 - 2\alpha_1L_1 - \alpha_1x}]$$
$$[1 - Ce^{-2\alpha_1L_1} - Ce^{-2\alpha_2L_2} - (1-2C^2)e^{-2\alpha_1L_1 - 2\alpha_2L_2}, \text{etc.}] \qquad [17]$$

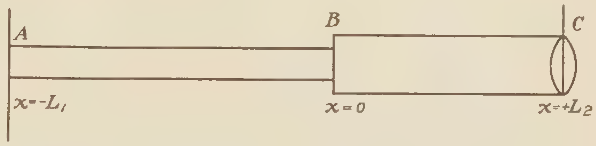

Fig. 5

$$P_2 = V_m\sqrt{W/Q_2}\,e^{-\alpha_2L_2 + \alpha_2x}[1 - Ce^{-2\alpha_2x} - Ce^{-2\alpha^2L_2}$$
$$+ C^2 e^{-2\alpha_2L_2 - 2\alpha_2x} - (1-C^2)e^{-2\alpha_1L_1 - 2\alpha_2x}$$
$$- (1-C^2)e^{-2\alpha_1L_1 - 2\alpha_2L_2}, \text{etc.}] \qquad [18]$$

where $\alpha_1 = p\sqrt{(WQ_1)}$ and $\alpha_2 = p\sqrt{WQ_2}$.

A study of Equations [17] and [18] determines that the initial wave $V_m\sqrt{(W/Q_2)} = P_m$ travels up the lower pipe with velocity a_2. When it reaches $x = 0$, the change in section, it continues through at velocity a_1, at the same time setting up a return wave of $-CP_m$, and an additional forward wave of $-CP_m$ so that the wave from B to A has magnitude $P_m(1-C)$. On reaching A, a return wave is set up $-P_m(1-C)$, which cancels the direct wave as far as B. This wave travels through B toward A un-

changed, but sets up an extra wave, $+C$ times itself, i.e., $-C(1 - C)P_m$ which goes both ways from B. On reaching the lower end, the wave sets up a return wave up the pipe exactly equal to itself, unchanged in sign. Thus, whenever a wave passes B, going up the pipe, an extra wave is started both ways of $-C$ times itself. A wave passing B going down the pipe sets up an extra wave $+C$ times itself. On reaching the intake the return wave is of opposite sign. On reaching the gate, the return wave is of the same sign. Equations [17] and [18] could be arranged to show this wave action more clearly and to proceed indefinitely in time. Note that these results check with the study of compound pipes by Billings, and coauthors.[2]

Example 3

In this example, consider the uniform pipe between two reservoirs, as shown in Fig. 6, when the gate valve is gradually closed at some intermediate point, the area being reduced uniformly with time until it reaches zero at time T_1. In this figure, the gate

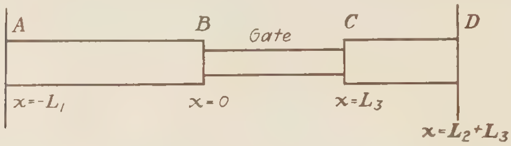

Fig. 6

is represented by the pipe length BC of variable area. The function F, shown in Fig. 7, which increases uniformly from 0 to F_m in time T_1 and remains constant at F_m, is represented by the operational form

$$F = (F_m/T_1p)(1 - e^{-T_1p})$$

This is evident by expanding

$$\frac{1}{T_1p} - \frac{e^{-T_1p}}{T_1p} = \frac{t}{T_1}\left(-\frac{t - T_1}{T_1} \text{ for } t > T_1\right) \ldots\ldots [19]$$

since e^{-T_1p} is zero up to $t = T_1$, and unity thereafter.

Since this function denotes the reduction in area of the gate, the expression for the gate area is

$$\frac{D_1^2 - D_3^2}{D_1^2} = \frac{1 - e^{-T_1p}}{T_1p} \ldots\ldots [20]$$

or, for convenience

$$\frac{D_3^2}{D_1^2} = 1 - \frac{1 - e^{-T_1p}}{T_1p} = 1 - F_3 \ldots\ldots [21]$$

We assume also for convenience that the wave velocity through the gate is the same as that in the main pipe. Our example then reduces to a pipe having three sections, the upper and lower ones constant and the center, of length L_3, variable in area. Using subscripts 1 and 2 for upper and lower sections, and 3 for the gate, the operational expressions are

$$P_n = A_n \cosh \alpha x + B_n \sinh \alpha x \ldots\ldots [22]$$

and

$$V_n = -\sqrt{Q/W}\,(A_n \sinh \alpha x + B_n \cosh \alpha x) \ldots\ldots [23]$$

where $n = 1$, 2, or 3 depending on the section, and $\alpha = p\sqrt{QW}$. Conditions of flow during the surge are $P_1 = 0$ at $x = -L_1$; $P_2 = 0$ at $x = L_2 + L_3$; $P_1 = P_3$ at $x = 0$; $P_2 = P_3$ at $x = L_3$; $(V_m + V_1)D_1^2 = (V_m + V_3)D_3^2$ at $x = 0$; and $(V_m + V_2)D_2^2 = (V_m + V_3)D_3^2$ at $x = L_3$.

In the last two conditions $(V_m + V_1)$ = actual velocity in upper section, and similarly for $(V_m + V_2)$ and $(V_m + V_3)$ since V_n = imposed surge velocity. Substitution and solution for the arbitrary constants leads to the six operational expressions for the pressures and velocities. Those for only the upper section are

$$P_1 = V_m\sqrt{W/Q}\,F_3 \sinh \alpha(L_1 + x)Y \ldots\ldots [24]$$

$$V_1 = -V_mF_3 \cosh \alpha(L_1 + x)Y \ldots\ldots [25]$$

where

$$Y = [\sinh \alpha(L_2 + L_3) - \sinh \alpha L_2 - F_3 \sinh \alpha L_2(\cosh \alpha L_3 - 1)] \div \{[\sinh \alpha(L_1 + L_2 + L_3) - F_3[2 \sinh \alpha L_1 \sinh \alpha L_2 \sinh \alpha L_3 + \cosh \alpha L_3 \sinh \alpha(L_1 + L_2)] + F_3^2 \sinh \alpha L_1 \sinh \alpha L_2 \sinh \alpha L_3]\}$$

To simplify the discussion, assume the gate to be at the lower end, that is, $L_2 = 0$, whence

$$P_1 = \frac{V_m\sqrt{W/Q}\,F_3 \sinh \alpha(L_1 + x) \sinh \alpha L_3}{\sinh \alpha(L_1 + L_3) - F_3 \sinh \alpha L_1 \cosh \alpha L_3} \ldots\ldots [26]$$

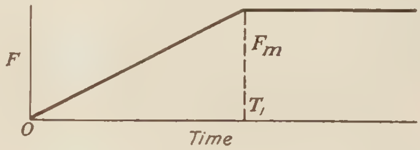

Fig. 7

It is evident that reduction of L_3 to zero reduces this operator to zero. The reason for this is that the waves which are reflected from the lower end of the gate cancel out the waves which proceed upstream from the upper end of the gate. Neglecting friction losses, the gradual closing of a gate between two reservoirs has little effect on the pressures according to the elastic formulas. The reduction in velocity causes the pressure rise, but this pressure rise cannot be used to increase the velocity through the gate. The usual practice of assuming the velocity through the gate as varying with $\sqrt{H_0 + H_R}$ where H_0 = normal head (i.e., head absorbed by friction) and H_R = pressure rise, is not based on elastic properties, but is an approximation partially based on friction losses. Here it is virtually assumed that the total friction head is normally available to give a discharge through the gate, i.e., V (gate) = constant $\sqrt{2gH_0}$ and that the pressure rise increases this available head. However, the H_0 loss is distributed all along the pipe, and the pressure rise H_R does not extend over the full pipe length except at one instant.

Example 4

In this example consider gate closure at the lower end of a simple pipe line discharging into the air, the area being reduced linearly to zero in time T_1; friction to be neglected. This is similar to example 3 but with $L_2 = 0$ and gate conditions such as to give no reflections of waves. To obtain this, it may be assumed that the gate length L_3 is infinite and, on this assumption, it will be necessary to change the form of the operational expressions for P_3 and V_3 to

$$P_3 = \frac{A_3 + B_3}{2} e^{\alpha x_3} + \frac{A_3 - B_3}{2} e^{-\alpha x_3} \ldots\ldots [27]$$

$$V_3 = -\sqrt{QW}\left[\frac{A_3 + B_3}{2} e^{\alpha x_3} - \frac{A_3 - B_3}{2} e^{-\alpha x_3}\right] \ldots [28]$$

[2] "High-Head Penstock Design," by A. W. K. Billings, O. H. Dodkin, F. Knapp, and A. Santos, First Water-Hammer Symposium, A.S.M.E., 29 West 39th Street, New York, N. Y., 1933, pp. 29–63.

The surge conditions are $P_1 = 0$ at $x = -L_1$; $P_1 = P_3$ at $x = 0$; $(V_1 + V_m)D_1^2 = (V_3 + V_m)D_3^2$ at $x = 0$; and P_3 is finite for $x = \infty$. The last condition is the only one that is new (to replace $P_3 = 0$ at $x = L_3$) and it gives $A_3 = -B_3$. Substituting and reducing, there results

$$P_1 = V_m \sqrt{\frac{W}{Q}} \frac{F_3 \sinh \alpha(L_1 + x)}{\cosh \alpha L_1 + (1 - F_3) \sinh \alpha L_1} \ldots [29]$$

and similarly for V_1. Transforming these hyperbolic functions to exponentials, this becomes

$$P_1 = V_m \sqrt{\frac{W}{Q}} \left[e^{\alpha x} - e^{-\alpha(2L_1+x)} \right] \left[\frac{F_3}{2 - F_3} \right] \left[1 + \frac{F_3}{2 - F_3} e^{-2\alpha L_1} \right]^{-1} \ldots [30]$$

From Equation [30] it may be seen that the direct pressure wave is $V_m \sqrt{(W/Q)}\ [F_3/(2 - F_3)]$, and each reflected wave at the gate is $+F_3/(2 - F_3)$ times the preceding wave from which it springs, or $-F_3/(2 - F_3)$ times the preceding outgoing wave. Within the time 0 to T_1, we may replace F_3 by $1/T_1p$. This gives the magnitude of the first direct wave as $V_m\sqrt{W/Q}$ $[1/(2T_1p - 1)]$ which expands to $[e^{t/2T_1} - 1]V_m \sqrt{W/Q}$. If there is no interference with returning or reflected waves, the pressure rise will therefore be $V_m \sqrt{W/Q}(\sqrt{e} - 1)$ or 0.649 $V_m\sqrt{W/Q}$ at $t = T_1$ after passing a given section, or 65 per cent of the pressure rise for instantaneous gate closure. The first reflected wave at the gate (at $t = 2L_1/a$) is $-V_m \sqrt{(W/Q)}[1/(2T_1p - 1)]^2$ which expands to

$$-V_m \sqrt{\frac{W}{Q}} \left[\frac{1}{\underline{|2}} \left(\frac{t}{2T_1} \right)^2 + \frac{2}{\underline{|3}} \left(\frac{t}{2T_1} \right)^3 + \text{, etc.} \right]$$

$$= -V_m \sqrt{\frac{W}{Q}} \left[1 - \left(1 - \frac{t}{2T_1} \right) e^{t/2T_1} \right] \ldots [31]$$

This analysis checks well with example 3. Using a finite L_3, it is found that the initial wave setting out from the upper end of the gate (not interfered with by reflections at $x = L_3$) has the magnitude $V_m \sqrt{(W/Q)}\ [F_3/(2 - F_3)]$. The use of Loewy's wave plan simplifies the study of the various waves and their reflections in the case of a pipe having two or more sections, or having a gate of appreciable length.

The assumption made in example 4 of an infinite length of gate, or an infinite length of pipe of reduced area with no friction losses and a smooth transition between the main pipe and the gate, cannot be considered as corresponding to true orifice conditions, but merely insures that no waves are reflected back from the lower end. The expression for pressures in the gate section may be determined as identical with that for the main pipe except for the delayed time factor, and the pressure does not diminish with distance from the main pipe. The use of a gate characteristic $Q_3 = \infty$ (corresponding to the modulus of elasticity $E_3 = 0$) would also simulate orifice conditions, but this cannot be assumed for the whole gate section as it would reduce the expression for P_1 to zero. Thus, it appears that the theory outlined in this paper, based upon frictionless flow, is inapplicable in a practical sense to flow at orifices, and gives only the theoretical pressure rise for a uniformly decreasing cross-sectional area of the lower end of a smooth pipe, neglecting wave reflections at the lower end.

In all such problems of large pressure and velocity changes, the friction factor and factors due to such conditions as non-uniform flow or eddies should not be neglected. Each of the hydraulic machines in such a network has its peculiar characteristics which must be accounted for, and to increase the accuracy of the solutions of these problems more detailed study must be given to the performance of these machines (e.g., valves of various types, turbines, pumps, and motors). For instance, in a relatively long pipe line where the friction-head loss at a certain regulated discharge is distributed along the pipe and where the discharge itself is perhaps regulated by some other device such as a turbine, it seems illogical to assume the velocity through the gate under steady conditions as varying with the root of the net effective head H_0, and that during the surges the velocity through the gate varies as the square root of (H_0 + pressure rise at gate). Usually a fully open gate offers little resistance to the flow, whereas a partly closed gate exerts a resistance depending greatly on its design.

The problem could be attacked by assuming that the whole system is made up of a series of hydraulic units interconnected by lengths of pipe line. Each unit has its characteristics which may be determined for the whole range of operating conditions (usually by test). Also, the elastic properties of the pipe lines permit the fairly accurate determination of pressure and velocity changes under varying conditions at their extremities. A correlation of these two—unit characteristics and pipe-line characteristics—based on the known causes of the surge, originating in one or more of the hydraulic units, should lead to a solution. An example will illustrate this.

Example 4a

In this example, consider a uniform pipe line between two reservoirs with the gate at some intermediate point. The operational expressions for pressure and velocity in the upper and lower pipe lengths P_1, V_1, and P_2, V_2, respectively, may be written either neglecting or allowing for friction. The gate opening may be designated by G, which varies from 0 to 1, and may be based on any convenient unit, such as angular turn or area of opening. At each gate opening G an empirical law of discharge through the gate may be assumed, such as $(P_1' - P_2') = \phi(G)V_3^2$ where P_1' and P_2' are the actual pressures at the upper and lower faces of the gate, V_3 is the equivalent velocity of discharge in the main pipe, and $\phi(G)$ is a function of the gate opening to take care of the characteristics of the particular gate. Knowing the law of closure of the gate $G = f(t)$, a function of time, then $P_1' - P_2' = \phi[f(t)]V_3^2 = \psi(t)V_3^2$, which is the condition imposed by the gate closure. Here, $P_1' = (P_1 + P_0)$ and $P_2' = (P_2 + P_0)$ assuming normal pressure at gate to be P_0. Also, $V_3 = V_m + V_1 = V_m + V_2$, where V_m = normal velocity in main pipe. Thus, we have a relation between P_2, P_1, and (V_1 or V_2) at the gate. The other three necessary conditions for surge flow are $P_1 = 0$ at upper end, $P_2 = 0$ at lower end, and $V_1 = V_2$ at the gate. These four conditions will give the arbitrary constants for the pipe equations. If the function $\psi(t)$ can be made simple, the operational solution is also simplified.

Example 5

In this example, consider a simple pipe line with instantaneous gate closure at the lower end, allowing for friction. The differential equations are

$$\frac{\partial P}{\partial x} + FV + W \frac{\partial V}{\partial t} = 0 \ldots\ldots\ldots\ldots [32]$$

and

$$\frac{\partial V}{\partial x} + Q \frac{\partial P}{\partial t} = 0 \ldots\ldots\ldots\ldots [33]$$

where $F = (2fW/D)(k_fV_m)$, representing friction. From Equations [32] and [33]

$$\frac{\partial^2 P}{\partial x^2} + (Wp + F)\frac{\partial V}{\partial x} = 0 \ldots\ldots\ldots\ldots [34]$$

or

$$\frac{\partial^2 P}{\partial x^2} = Qp(Wp + F)P \ldots\ldots\ldots\ldots\ldots [35]$$

Equation [35] has the operational solution

$$P = A \cosh \alpha x + B \sinh \alpha x \ldots\ldots\ldots\ldots [36]$$

where α is the positive root $\sqrt{[Qp(Wp + F)]}$.

The surge flow conditions are $P = 0$ at $x = 0$ (upper end) and $V = -V_m$ at $x = L$ (lower end). These give

$$A = 0 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [37]$$

$$B = \frac{V_m \sqrt{[(Wp + F)/Qp]}}{\cosh \alpha L} \ldots\ldots\ldots\ldots [38]$$

and the operational solutions become

$$P = V_m \sqrt{\frac{Wp + F}{Qp}} \frac{\sinh \alpha x}{\cosh \alpha L} \ldots\ldots\ldots\ldots [39]$$

and

$$V = -V_m \frac{\cosh \alpha x}{\cosh \alpha L} \ldots\ldots\ldots\ldots\ldots [40]$$

Only the pressure surges are studied here since the operators sinh αx and cosh αL require much more manipulation than has been required in the foregoing examples and the detailed work will not be shown. However, it may be checked by reference to standard books and articles on operational calculus; in particular, Bush's "Operational Circuit Analysis," pp. 216 et seq (1)[3] and Campbell and Foster's list of operators (3).

The final form for the initial pressure surge is

$$P_1 = V_m \sqrt{\frac{W}{Q}}\, e^{-F_0 t} \Big[1 + (2 + \tfrac{1}{2}T_1F_0)F_0(t - T_1) + \left(\frac{5}{2} + T_1F_0 + \frac{1}{8}T_1^2F_0^2\right)\frac{F_0^2}{\lfloor 2}(t - T_1)^2 +, \text{etc.}\Big] \ldots [41]$$

where $F_0 = F/2W$, and T_1 is the time required for the wave to reach section $= (L - x)/a$. The return surge from the intake P_2 is minus this expression for P_1 with $T_2 = (L + x)/a$ substituted for T_1; similarly, the second wave from the gate to the intake P_3 is the same as wave P_2 with $T_3 = (3L - x)/a$ substituted for T_2, and so on. None of these waves is effective until $t = T_1$, T_2, T_3, etc.

A study of the pressure surge at the gate will prove instructive. Here $T_1 = 0$; $T_2 = T_3 = 2L/a$; $T_4 = T_5 = 4L/a$; etc. The first pressure wave is $P_1 = V_m \sqrt{(W/Q)}\, e^{-F_0 t}$, which is the maximum surge multiplied by the damping factor $e^{-F_0 t}$. This holds until $t = 2L/a$. After that the pressure is augmented by the P_2 and P_3 surges, etc.

In connection with the friction factor k_f it is to be noted that the differential equations are solved for the imposed surges and not for the actual pressures and velocities along the pipe. It has been found that the imposed velocities for sudden gate closure, neglecting friction, fluctuate between $-V_m$, $-2V_m$, and zero. Fig. 2 shows that with an imposed velocity V_1 the actual friction in the pipe is given by CD so that the regained friction head is given by DE. Thus, for very small velocity surges we should have double the friction factor, i.e., $k_f = 2$, since the friction curve has twice the slope at the point A as the line OA. For imposed velocities 0, $-V_m$, and $-2V_m$, the factor $k_f = 1$ is evidently correct.

Particular cases, in which the friction factor $F_0\mu = 0.4$, 0.2, and 0.05 have been examined by the author, and the surge

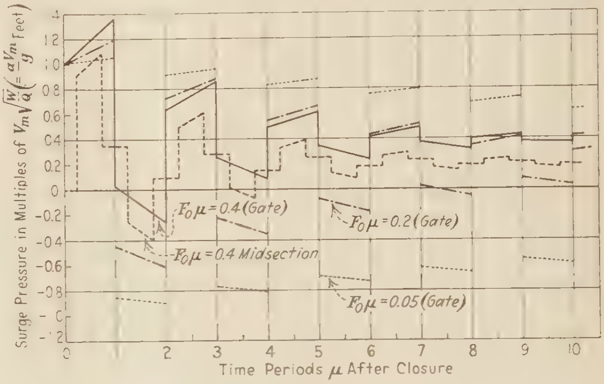

Fig. 8

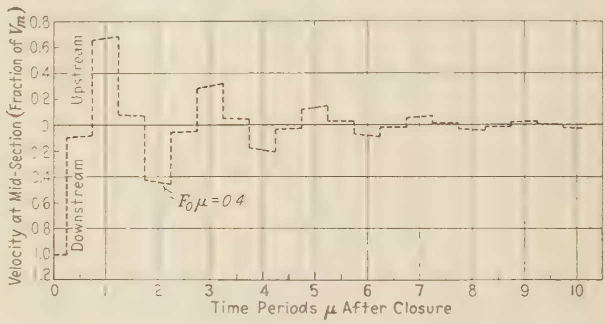

Fig. 9

pressure and velocities obtained are plotted in Figs. 8 and 9. Since the factor $F_0\mu$ is found to be a criterion of similarity of surge conditions, it seems logical to designate it by a special name, such as "friction modulus." Referring to Fig. 9, it is evident that for velocities downstream friction allowance according to the linear law is deficient, and for the velocities upstream this approximation for friction is excessive. A more accurate determination of the surge pressure may be obtained by a correlated study of the friction, pressure, and velocity charts shown in Figs. 2, 8, and 9, respectively.

Example 6

In this example, consider the sudden closing of the gate in a simple pipe line which has a surge tank at its lower end; friction to be neglected. Let L = length of pipe of diameter D_1 and area A_1. Also, let the initial head in the surge tank be H_0 above the center of the pipe, and the area of the tank $= A_2 = SA_1$. Then the force-momentum condition imposed on the surge tank leads to the differential equation

$$P_2 - wH_2 = wH_2 \frac{dV_2}{dt} + wV_2 \frac{dH_2}{dt} \ldots\ldots\ldots [42]$$

[3] Numbers in parentheses refer to the Bibliography at the end of the paper.

and the condition for continuity of flow gives

$$V_2 = S\frac{dH_2}{dt} \qquad [43]$$

where the subscript 2 denotes the surge-tank entrance to the main pipe line, the diameter of which entrance is assumed equal to the diameter of the pipe line. For the main pipe

$$P_1 = M \cosh \alpha x + N \sinh \alpha x \qquad [44]$$

and

$$V_1 = -\sqrt{(Q/W)}\ (M \sinh \alpha x + N \cosh \alpha x) \qquad [45]$$

where $\alpha = p\sqrt{(QW)}$.

Now the conditions of surge are $P_2 = P_1$ at $x = 0$ (lower end of the pipe line) and $P_1 = 0$ at $x = -L_1$ (upper end of the pipe line). Imposed surge velocities $V_1 - V_2 = -V_m$ at $x = 0$. Unfortunately, the two terms on the right of the expression for P_2, Equation [42], are not linear. As an approximation it is possible to neglect the dH_2/dt term and replace $wH_2(dV_2/dt)$ with $w(k_sH_0)(dV_2/dt)$. Certain solutions offered by earlier theories assume $k_s = 1$. In any particular example, it may be possible to adjust this coefficient after a first approximation has been obtained.

The surge-tank operational equation reduces to

$$P_2 = wH_2 + wk_sH_0(dV_2/dt) = [(w/Sp) + wk_sH_0p]V_2 \qquad [46]$$

Using the surge flow conditions and solving for M, N, and V_2, there results the operational solution for the pressure at the gate or base of the tank

$$P_2 = \frac{V_m\sqrt{(W/Q)}\sinh \alpha L_1}{\cosh \alpha L_1 + \sqrt{\frac{W}{Q}}\,\frac{p \sinh \alpha L_1}{w[k_sH_0p^2 + (1/S)]}} \qquad [47]$$

It should be noted that it is the right-hand expression in the denominator of Equation [47] which distinguishes example 6 from example 1 (the simple pipe line without a surge tank). This last expression will expand into a series of reflected waves and the general analyses can thus be secured.

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Mechanical Processing of Vegetable Oils

By W. WADE MOSS, JR.,[1] NEW YORK, N. Y.

The author discusses the function of unit-operation engineering as applied to the mechanical processing of vegetable oil, and although he does not attempt to make a comprehensive study of oil processing, he shows the importance of certain basic factors required for the efficient refining of edible oils. He elaborates on recent work at the University of Tennessee on the use of pressure cooking at higher temperatures than normally used, and compares the advantages and disadvantages of varying cooking pressure, temperature, and time.

The paper includes a discussion of different means of processing oils by hydraulic expeller expression and extraction by solvent methods. A detailed description is given of a hydraulic press of the newer pot type which combines filter action, press action, and automatic ejection.

The author points out that the moisture in the press cake varies according to the nature of the oil, and that by maintaining the proper percentage of moisture the cakes can be prevented from pasting, thereby permitting easy removal from the press cloths. He gives data relative to bleaching admixtures of char and fuller's earth and makes recommendations as to the proper bleaching compounds to use in order to obtain the best result when processing the oils.

The author also discusses the advantage and disadvantages of impeller and piston-type vacuum pumps, and thermocompressors, and recommends the use of thermocompressors which are able to maintain a vacuum of 29.5 in. Hg in the vapor kettles while deodorizing the oils.

THE MECHANICAL processing of vegetable oils is an old art and much of the technique employed during past generations is still used today. With the increased consumption of vegetable oils, especially the wider scope of the refined vegetable oils, has come a demand for higher quality and cheaper methods of processing; consequently, many outmoded unit operations have been replaced by highly efficient improvements. Where once the vegetable-oil processing industry functioned as an art, it is now fast becoming a highly organized, systematic science. Much of the skill now exercised in the vegetable-oil processing field is the result of borrowed unit operations from other fields such as the chemical and food industries.

This paper purports to set forth some of these borrowed unit operations and to indicate in what manner these advances have benefited the oil-processing industry. It hopes to illustrate the futility of competition by those who cling to ancient methods of vegetable-oil processing. The author intends primarily to present improved unit operations as individual units without consideration of their respective interrelation in a composite flow sheet. The chemical or the mechanical engineer, it is hoped, will find one or more of these unit operations which he may adapt, in part or in toto, to his particular process.

The introduction of unit-operation engineering into the processing of vegetable oils came gradually. Extensive research in the field of chemical engineering and its classification into highly specialized unit operations such as, for example, distillation, adsorption and extraction, and filtration created new principles and new technique. The greatest evidence of the influence of improved chemical engineering in the vegetable-oil processing is demonstrated in distillation or, as it is called in oil-processing parlance, deodorization operation. Noteworthy economies have been effected in the deodorization technique by the introduction of high-velocity vapor vacuum systems utilizing thermocompressors which were considered commercially impractical by the vegetable-oil men until they were proved feasible in the petroleum-refining industry. From the sugar industry came the improvement in the use of bleaching compounds.

Mention is made briefly of those unit operations which have undergone little change, or which, in the light of present knowledge, appear to be efficient in operation with a minimum of cost. It is suggested that a concerted study, beyond the scope of this paper may assist the vegetable-oil-processing engineer to obtain greater uniformity of product, standardization of production, lower processing costs, and greater efficiency.

Pretreatment of the Seed

Rolling. The final quality of the processed vegetable oil depends to a large extent upon the preliminary treatment given the seeds prior to their expression or solvent extraction. Two factors must be observed. First, a maximum quality of oil of the best attainable grade must be recovered, and, second, the meal must not be so badly injured that it becomes unsalable. The methods employed in the first of the preliminary treatments of the seeds is more or less universal; the seeds are rolled in a specially designed apparatus which mash them into a flake in order to rupture the oil-bearing cells, breaking the enclosing membrane to allow the oil to flow forth unimpeded. Unless some means is used to prevent it, the subsequent yield of oil is low, regardless of other treatments. In terms of standard flow sheets, as a unit operation, rolling is more satisfactory than any other method, such as grinding, milling, or macerating. For any given rolling machine, the number of rolls, size of rolls, and space allowed between rolls for pressing the seeds into flakes of a desired thickness are matters of specification. For example, a machine can be purchased which consists of five 42 × 14-in.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

steel rolls set so as to reduce the seeds to flakes of approximately 0.007 in. thick.

Cooking. After the seeds are rolled into flakes, the treatment to rupture the cell walls is continued by cooking the flakes. Care must be exercised in this treatment, because too low a cooking temperature will not release the oil, while too great a temperature will result in an off quality of oil and injure the meal residue after expression. The cooking temperatures are functions of the time; a cooking temperature of approximately 230 F requires between 1 and 1.75 hr. While it is possible to shorten the time of cooking by substantially raising the temperature, the injury done to the nitrogen factor in the meal does not justify the time saved. During the cooking careful control of the moisture content of the seeds should be maintained. The importance of the proper percentage of moisture in the seed mass during expression is important to the quality of the oil and to the wear and tear upon the press cloths.

The cooking is usually done in open steam-heated tanks and agitation is necessary. In practice, this cooking is usually done at atmospheric pressure; however, the recent work carried on at the University of Tennessee[2] using pressure cooking has great possibilities. In this case a large number of experimental cookings were made at varying times and varying temperatures. The yields of oil have been reported as considerably higher and the quality of the oil equal, at least, to the best oil obtained when using atmospheric cookers. During these cookings it was possible to control the moisture within a close range by moisture-control equipment consisting of an auxiliary tank so constructed that either moisture could be added to or removed from the meats at will. The apparatus receives its supply of steam from a steam generator designed to admit either steam or cold water, thereby giving a reversible action. When it was necessary to remove some moisture from the meats, cold water was blown in and evaporated off into the measuring tank. Thus, the meats gave up this excess moisture, retaining within narrow limits the requisite quantity; no other known treatment gives such unusual quantitative control. When adding moisture, steam is blown in at a rate which permits the moistening of the meats to approximately 3 per cent plus or minus, according to the nature of the oil to be extracted. The method as a whole demonstrates a decided advance over any now in use.

The actual cooking tank as employed in the work at the engineering experiment station of the University of Tennessee[2] combined a mixer and an agitator in a steam-jacketed horizontal steam-tight tank. Access to the tank was gained through a valve-like door. The operating temperatures employed were 270 F under a pressure of 25 lb per sq in. with a cooking time of 15 min.

Extraction

Expression. The extraction of the oil from the treated seeds may be carried on by three methods; namely, expression by an Anderson expeller, by pressing in an hydraulic press, or by solvent extraction. In the case of solvent extraction, when used merely as a means of protecting proteins in the meal, a combination of pressing and subsequent solvent extraction is best. In this way, larger percentages of oil come out as good press oil, which is better for many purposes than solvent-extracted oil.

In Europe, the Anderson expeller is much more widely employed, while in the United States, the hydraulic method is favored. German edible oils are produced by the solvent-extraction system. On the contrary, in America pressure-extracted oil is given preference and solvent extraction is mainly used in cases in which it is desired to preserve the proteins in the meal for other than food purposes. In the preparation of meal for plastics, solvent extraction is essential in order to remove the 9 to 10 per cent of residue oil remaining in the press cake after ordinary pressing.

Many types of hydraulic-pressure equipment are available, the design of the presses varying only in minor details according to the manufacturer. This makes possible a standard technique throughout the industry for the preliminary treatment of the seeds, since such treatment can be followed without regard to the type of press that is to be employed.

The new Carver vertical hydraulic press which exerts a pressure of 5000 lb per sq in., offers certain advantages in simplicity of operation and performs a combination of functions in one cycle. As an example, a Carver five-pot press can handle 50 lb of meal per pot or 250 lb of meal per cycle; at 3 cycles per hr, the hourly production becomes 750 lb. In one day of 10 hr, two such presses can handle one half a car of peanuts, for example. The operation of this type of press is so simple that one man can handle two presses; the oil is pressed out through cylindrical filter plates in the pots, and the meal is ejected from the pot. Then the next cycle follows.

An even greater improvement is the horizontal hydraulic filter press which is designed to exert pressures of from 1000 to 5000 lb per sq in. The material is pumped into the press through a feed manifold. Then, operating in cycles, the pressing begins, the oil is filtered, and as the filtrate runs off it is caught by a deflector. The cakes, which are now ready to be discharged, drop out of position and are picked up by a conveyor and removed. The press is returned to position and is ready for the next cycle. In operation the cycles respond to valve control thus minimizing labor and handling of materials.

Solvent Extraction. Modern solvent extraction involves many secret trade processes. In studying the methods used, only the barest basic principles are general in adoption. Such solvents as hexane and alcohol, and the hydrocarbons are used, although preference is given the hexane-alcohol treatment. In the extraction, many patented types of apparatus are employed and many variations of basic adsorption, extraction, and reflux technique are used. Those manufacturers in the United States who use solvent extraction are doing so, for the greater part, with a definite purpose in mind; for instance, the Ford system for the extraction of soy-beans extracts the defatted soy-meal in an especially designed apparatus (Edison Institute) in order to obtain proteins from the meal.

In the case of expression technique, the cooking resorted to and discussed here involves temperatures which destroy the heat-sensitive proteins of the vegetable seeds or of the nuts. Solvent extraction alone enables the use of an extraction temperature which prevents any injury to the proteins.

The color of the oil resulting from the extraction is slightly different from that obtained by expression. Much preference is given pressed oil for use as an edible oil. Solvent extraction is recommended only in cases where the meal is to be preserved.

Lecithin

During recent years, processors of vegetable oils have realized the value of recovering the phospholipides from vegetable oils. Solvent extraction is thus far the only satisfactory means of recovering these compounds. However, contrary to popular belief, it is found that a large quantity of the phospholipides come over with expressed oil, since it is extremely difficult to isolate them from expressed oils.

During the days when solvent extraction was first becoming accepted in the industry, it was noticed that a long period of settling was necessary in order to remove a gummy, mucilagi-

[2] "Cotton Seed Oil Pressing Costs Cut," by C. A. Perkins and R. B. Taylor, *Food Industries*, vol. 9, August, 1937, p. 435.

nous substance from the solvent-extracted oil. Thus, the oil was allowed to settle for days until it became clear. Today this gummy mass is by far the largest source of commercial phospholipides which are called "lecithin."

Commercial lecithin is now extracted from the solvent-recovered oil by centrifuging. The compound is an admixture of oils, free fatty acids, a protein group (amido), and combined phosphorus. The article as known in commerce is not pure lecithin, but contains a large percentage of carbohydrates (up to 10 per cent). These substances have properties of the fatty oils and many of the proteins.

After the lecithin is isolated, it is dried in vacua whereupon, it becomes a sticky, gummy, and reddish brown mass. As such it is used in the rubber and leather industry, and in foods as an antioxidant. Only about 2 per cent exists in the vegetable seed, of which about 20 to 30 per cent is recoverable.

The foregoing is mentioned as an explanation of the value of solvent extraction. Except for this purpose, the preference for expression technique is extant in the United States.

Neutralizing

Neutralizing vegetable-oil fatty acids is essentially a simple operation fraught with many difficulties unless skillfully manipulated. The character of the oil under treatment governs to a large extent the details of the process. Assuming that the processor has fitted the reaction conditions to the characteristics of the oil, the following suggestions can be successfully used: The treatment of the oil with soda causes a "break" in the oil, and free fatty acids as flocculent particles of soap gather together with water and fall to the bottom of the tank as foots; a sufficient quantity of soda is used to cause the formation of the soap with the total of the free fatty acids and an excess to serve as a deflocculent and cause greater and more rapid precipitation of the soap flakes. Frequently, the oil contains extracted carbohydrates from the meal. The presence of excess soda causes the solution of these sugars, especially the pentose sugars. These sugars tend to form gummy products with the precipitated soap, and fall to the bottom of the tank. Careful regulation of this operation often results in a good cleaning of the oil.

The time, the temperature, and the degree of agitation of the oil during neutralizing are specific for a given oil.

After the saponification has been completed, the oil and the soap are run into a large settling tank and the soap is settled out of the oil; however, it may also be salted out. The formation of emulsions can be avoided during neutralization by sufficiently strengthening the soda solution. When emulsions form, a loss of neutral oil occurs which assists in making the emulsion as well as increasing the time of settling out of the foots. Experience alone can provide the conditions at which the consistency of the soap stock is such that a uniform settling time results together with the maximum degree of neutralization and minimum loss of neutral oil.

Bleaching

The use of bleaching carbon greatly facilitates the purification of the oil. The removal of colors and other absorbed impurities is accomplished by the use of activated carbon; although best results can be obtained by the addition of a bleaching earth. The use of two-component mixtures reduces the quantity of purifying agent required. Fuller's earth has long been recognized as the standard absorbent in the vegetable-oil industry. However, it does not have the adsorptive properties attributed to activated charcoal, but because of its cheapness it has been used by the industry in previous years as the sole bleaching agent. The introduction of activated carbon failed to supersede entirely the bleaching earths not only because of the comparatively high price of activated charcoal, but also because these earths are fairly satisfactory in removing the yellows from the vegetable oils even though they do not remove the red shades. Compared with activated carbon, a higher percentage of earths than of activated charcoal is required to effect the same degree of bleaching.

Many factors enter into the bleaching of oils with earths or chars or combinations of both. Mechanical mixing and subsequent extraction is not sufficient. Consideration must be given to the type of bleaching agent which satisfies the nature of the oil being bleached. Oils differ greatly as to physical properties. The colloidal constituents and other impurities vary between wide limits. The treatment of the bleach compounds themselves deserves consideration because during purification many of their physical properties are changed materially. This change together with the wide variance in characteristics of the vegetable oils presents a problem that requires a long series of tests to determine the proper bleach for a given oil.

In the decolorization of these oils, in which there is suspended colloidal matter such as proteins, the rate of decolorization as well as the efficiency of the decolorizing carbon or earth will vary with the pH. Amphoteric substances such as protein are adsorbed at or near their isoelectric point, at which point these substances show neither acidic nor basic properties.

Fuller's earth functions as a bleach more or less in ratio to its moisture content; however, its efficiency varies according to its powers of colloidal dispersion. The less the percentage of moisture, the greater the bleaching power. Electrokinetics enters into the selection of the proper bleach compound; assuming that the earths follow the laws of colloid adsorption, it will be found that the definite effect of one specified oil upon the bleach also follows the potential existing around the particles of bleach mix. Considering that fuller's earth, for instance, carries an electronegative charge, it would be useless with a colloidal impurity in the oil when such impurity bears an electronegative charge. Since some of the coloring matter in the oils existing as finely divided matter is electropositively charged, the fuller's earth, we assume, will adsorb the color. It is known that chars will carry different charges in the same body and therefore will adsorb some of both the electronegative and the electropositive colloids. This explains, to a degree the greater efficiency of the activated char when used alone with vegetable oils. Over a period of several years, the author tested the use of char alone with coconut oil and, from the final deductions which indicated that the use of char alone was satisfactory, it became standard practice in a large coconut-oil processing plant. However, with peanut oil, it was found that char did not give maximum efficiency. In this case, a larger quantity of char was needed and tests revealed that a color selectivity resulted. With the use of an admixture of 90 per cent fuller's earth and 10 per cent char, on the basis of 1–2 per cent of the foregoing mixture per charge of oil, the decolorization reached its maximum efficiency.

Many processors during the past ten years have used plaster or other agents to adsorb water from the oil to prevent dilution of the fuller's earth with excess moisture. Such agents lessen the activity of the earths or other bleach compounds, and aid in attaining the proper mix of bleach compounds suited to a particular oil, thereby reducing the amount of bleach compound required. Inasmuch as the refining loss is greatest because of adsorption of oil during the bleaching, it is essential that a compound be found that will reduce this loss to a minimum. Only by experiment is it possible to develop a bleach for a specific oil which gives maximum adsorption with minimum refining loss.

It has been found that oils can be bleached effectively under vacuum conditions, since, under such conditions, the adsorbing ability of the earth is increased and makes it possible to remove

moisture well below the point of too great desiccation. The relative adsorptive value of an earth reaches its maximum at a point between complete desiccation and partial desiccation; too great desiccation inactivates an earth. The only argument in favor of the use of vacuum in bleaching oils is the fact that it lowers the refining loss by requiring a lesser amount of bleach compound for a given amount of work.

No one can set forth a rigid infallible schedule for the processing of any one oil, stating the proper type of bleach and processing condition (that is, vacuum or atmospheric) which should be used; the engineer must resort to tests for solving each case individually, taking into consideration the general principles involved as discussed previously. The engineer is also cautioned to provide means of preventing the development of an earth-like flavor in the oil as a result of using the bleach compound.

The use of filter aids in the filtration of the bleached oil is a matter of opinion, but they cannot be selected at random since the times and type of filtration to be used are involved. The quantity of filter aid to be used, if any, depends upon the percentage composition of bleach compound or bleach substance. When the chars are used alone, as in coconut oil, very little if any filter aid is required, although many processors even in this case prefer to use a filter aid. Then again there is always refining loss to consider. A filter cake may be considered as a series of large capillary tubes through which the oil passes in a straight line. Each of these capillary tubes presents electrokinetic phenomena which govern the speed of delivery of the oil and the relative retention of the filtering oil. Here again, comes the problem of selecting the proper technique suited only to the oil being processed. It is necessary that attention be given to both the type of filter aid and the technique of application. One very good method is to spread the filter aid over the surface of the cloths and thus form a filtration medium to serve as a foundation membrane. The remainder of the filter aid is admixed with the bulk of the oil.

The general technique employed in the bleaching is comparatively simple after the bleaching and filter media have been selected correctly. The oil in the bleach tank is heated with steam in steam coils and raised to the bleaching temperature of about 230 F when used at atmospheric pressure. When a vacuum is used, the corresponding temperature will vary with the inches of vacuum; the higher the vacuum, the more rapid and more complete will be the removal of water vapor, and the lower will be the temperature. This vacuum can be readily achieved by the thermocompressor type of vacuum equipment which is discussed later in this paper.

During a period of 20 to 30 min., agitation is applied to the mass and thorough mixing is obtained. When a test sample shows that there is no more bleaching action, the mixture is pressed out in a filter press of the plate-and-frame type; during the pressing steam and air should be blown through the press to remove as much oil as possible and to dry and harden the cake. The oil should be kept away from air as much as possible, because it tends to darken the oil, this being especially true for some oils more so than others; however, all vegetable oils tend to darken with prolonged exposure to heat and air. Should the oil on first filtration appear cloudy, it is advisable to repress.

Deodorizing

Greater efficiency and low operating cost in deodorizing vegetable oil are possible by the use of the proper high-vacuum equipment, but in selecting, operating, and maintaining such equipment, it is necessary to understand those principles which under given circumstances have produced maximum results. It is possible to use either the piston-type, rotary-type, or the mercury-vapor-type vacuum pump, or to use the high-velocity vapor system of evacuation. These established means of obtaining high vacua are not synonymous and cannot be used interchangeably and still obtain production of comparable efficiency.

The deodorizing of vegetable oil is a separate distillation operation in which the highly volatile substances are admixed with the water vapor which forms a binary azeotropic mixture of a heterogeneous nature in a two-component-phase system.

In considering the relative efficacy of scrubbing or distillating steam, such factors as specific volume of the steam in the ratio to pressure, the velocity of the steam, and the temperature must be considered as functions of the weight of steam. In other words it is not the weight of the steam or the specific pressure of the steam that effects distillation in a given period but it is rather the specific volume as a function of steam velocity at any given moment that does the effective work. This simply means, for instance, that inasmuch as 1 lb of steam occupies a definite volume, with the introduction of variables such as temperature and pressure, the working condition of the steam is changed. In a low vacuum of say 25 in., the efficiency of the steam expansion by deodorization would be very low. Hence, the amount of steam necessary to volatilize a unit area of oil in the deodorization tank would be very large, and considerable steam would have to be passed through the tank. On the other hand, as the vacuum becomes greater, the amount of steam necessary to effect the deodorization becomes increasingly less and varies as the degree of vacuum. In other words, the steam under a very low vacuum must of necessity be added until the expansion of that given amount of steam in the deodorizer is equal to the ratio between the volume of steam introduced and the oil in the tank. The curve is not a constant function when the scrubbing steam is introduced at vacua over 29.5 in. Hg.

In order to illustrate the functional relationship between steam velocity and degrees of vacuum, let us consider a relative comparison between vacua 25.8 in. and 29.5 in. Hg, considering that 1 lb of steam occupies 26.79 cu ft at atmospheric pressure. In tracing the efficiency, we see that at 20.5 in. Hg the steam volume becomes 469.3 cu ft while at 25.5 in. the same weight of steam occupies a volume of approximately 3000 cu ft. Considering that water boils at 32 F at a vacuum of between 29.7 and 29.8 in. Hg, based on a 30-in. barometer, it is obvious that by maintaining a vacuum of 29.5 in., the ideal conditions for water vapor are almost reached. In the jump between steam at atmospheric pressure to steam at a vacuum of 28.5 in. Hg, the volume of steam increases 17 times. In other words the relative efficiency has already been increased by the expansion from 26.79 cu ft at atmospheric pressure to 469.3 cu ft at a vacuum of 25 in. Hg. Therefore, it would seem that while the ideal vacuum appears to be between 29.7 and 29.8 in. Hg, it is found that the efficiency curve varies abruptly during the last $1^1/_2$ in. of vacuum. Therefore, it is more expensive to attain the ideal vacuum, as we shall see in the following study of a commercial application of high vacuum. It is more practical to merely approximate the ideal vacuum by maintaining it at 29.5 in. Hg. This statement is made in consideration of mechanical efficiency possible with the vacuum-producing apparatus.

Therefore, considering that the range of high evacuation depends primarily on the tremendous increase of vapor at high vacuum, it is natural to contend that an apparatus should be used that is capable of gathering and dispersing high-velocity vapors. Or, to state the mechanics of high evacuation in a simplified form, let us say that density of any gas or vapor decreases in a definite rate with higher vacuum. The gas must flow at high velocity in any system of steam or other vapor in order to produce a high vacuum. It is essential to consider that gas with

a low density can obtain high vacuum because it can flow with less friction than a gas with a higher density. Now, considering the fact that the volume of steam increases as the vacuum increases, the cost of deodorization, it is found, varies accordingly.

Greater efficiency in the deodorization of vegetable oil has been made possible by the extension of high-vacuum thermocompressors known as steam-jet or steam-evacuator boosters. Low-vacuum redistillation provides an expensive method, while high-vacuum distillation by the use of steam jet and thermocompressors achieves deodorization of oil at a lower cost than formerly.

Steam jets are the logical means of obtaining high vacuum in commercial processes when there is an enormous amount of vapor to be handled. For this purpose of maintaining very high vacua and handling these large quantities of vapor, the thermocompressors and steam jets are more efficient than rotary or piston vacuum pumps. They are however, limited in maximum efficiency to a vacuum of approximately 29.5 in. Hg, in commercial application.

The mechanical function of these thermocompressors depends on the ratio between the velocity of incoming steam and the velocity of the vapor which leaves the deodorization kettle; that is, the flow of gas of low density tends to create high vacua which are necessary where a large quantity of steam must be handled.

The alternate cycle of expansion and extraction of high-velocity steam and the compression of the vapor requires no moving parts. The entire process depends upon the speed or velocity at which the steam enters the venturi throat of the steam-jet evacuator. Natural variations of the steam velocity, caused by variations of initial pressure, influence the degree of vacuum obtained inasmuch as the vacuum is dependent on the cyclic relationship of times per unit time that the vapor is withdrawn or sucked from the vacuum kettle and compressed. This is a function to the amount of steam at a given velocity that passes into the venturi throat in unit time. In the case of a vacuum pump of piston or rotary type, it is easy to see that mechanical limitations of the moving parts tend to make impractical the use of mechanical pumps for attaining a vacuum as high as 29.5 in. Hg. It is practically impossible to manufacture mechanical pumps that synchronize their movements with the high velocity of the liberated vapor; in other words, the enormous amounts of vapor leaving the vacuum kettle at such extremely high velocity would necessarily make difficult the practical production of machine parts to move at such a rate of speed. Also the cost of providing sufficient energy to move mechanical parts of piston or rotary vacuum pumps is too great.

The theory of high evacuation, based on the tremendous volume increase of vapor at high vacua, is illustrated by the following example of an actual installation in a small plant equipped with a single-stage thermocompressor with a 10-in. suction and an 8-in. discharge capable of maintaining a vacuum of 29.5 in. Hg. This plant deodorizes a 1000-gal batch of coconut oil in 3 hr, utilizing 300 lb of scrubbing steam per hr injected into the bottom of the deodorizer, and 110 lb of steam per hr for the steam jet used to maintain the vacuum; this makes a total of 410 lb of steam per hr for the unit, or 1230 lb of steam in the 3-hr for the 1000-gal batch of oil. A barometric surface condenser is used for condensing the steam from the thermocompressor; any air which leaks into the unit is removed from the condenser by a second steam jet the steam from which is usually condensed in the condenser circulating water after it leaves the condenser.

Many decided advantages are obtained by using multistage thermocompressors, because there are certain applications in which the operation proceeds under circumstances where the vapors are conducted ahead of the vacuum apparatus and must be removed. These multistage thermocompressors are essentially an expanded type of single-stage unit. In a three-stage unit, for example, the thermocompressor, as the first steam jet is called, is mounted on the vacuum kettle and pulls the vapors from the kettle and exhausts them into the first of the barometric condensers. A second steam jet mounted on the first condenser exhausts into a second condenser, and a third steam jet mounted on the second condenser exhaust into the third condenser. A fourth steam jet exhausts air from the unit, thus enabling it to maintain the vacuum; the steam from this condenser exhausts into condenser circulating water.

Running-In Characteristics of Some White-Metal Journal Bearings[1]

By S. A. McKee[2] and T. R. McKee,[3] Washington, D. C.

This paper describes an extension of the investigation of the running-in characteristics of journal bearings which was a part of the program of research on lubrication conducted at the National Bureau of Standards in cooperation with the Special Research Committee on Lubrication of The American Society of Mechanical Engineers. The results of tests on three kinds of white-metal bearings are given.

A four-bearing friction machine was used to determine the effect of progressive amounts of running in, upon the frictional characteristics of the bearings. The results are compared with those of previous investigations and all are in agreement in providing an indication that the major effect of running in is to reduce the friction losses at low values of ZN/P and to increase the permissible operating range. The results also give a comparison of the frictional and running-in characteristics of the various metals.

Early in the program of research on journal-bearing lubrication conducted at the National Bureau of Standards in cooperation with the Special Research Committee on Lubrication of The American Society of Mechanical Engineers, a study was made of the running-in characteristics of one kind of white-metal journal bearings. The results of these tests were given in a paper (1)[4] presented at the A.S.M.E. Annual Meeting in 1927. As the work progressed the study of running in was extended to include tests on bearings lined with three other kinds of white-metal alloys. The results of these tests are reported in this paper.

Method and Apparatus

The method and apparatus employed were the same as those used in the previous tests and are described in detail in the original report. The method involves the determination of the frictional characteristics of a given set of bearings for different degrees of running in. The usual graphical representation of the frictional characteristics (2) is obtained by plotting the coefficient of friction f against the generalized operating variable ZN/P where Z is the absolute viscosity of the lubricant in centipoises, N the speed of the journal in rpm, and P the pressure on the projected area of the bearing in lb per sq in. The effect of running in upon a given set of bearings is indicated graphically by the changes in the f versus ZN/P curve caused by various amounts of running in.

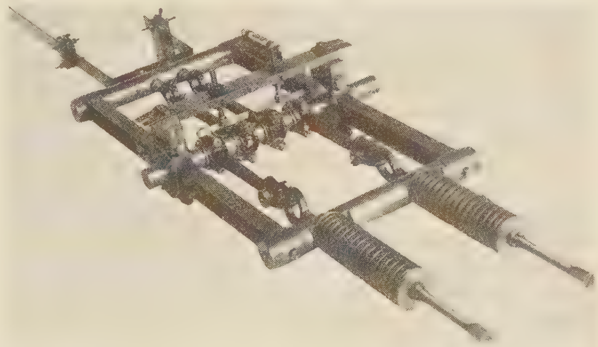

Fig. 1 Four-Bearing Friction Machine, Essential Parts

The friction data were obtained on the four-bearing friction machine shown in Fig. 1. The four test bearings operate on a steel shaft mounted in a lathe. The bearings are held in place by a frame provided with helical springs for applying the load. The frictional torque is determined by measuring the moment necessary to prevent the frame from rotating with the shaft. Oil is fed under a pressure of about 1.5 lb per sq in. from a glass reservoir through rubber tubing to the unloaded side of each bearing. Temperature rise is minimized by circulating water at room temperature through the hollow test shaft. The operating temperature is measured by the use of copper-constantan thermocouples which are mounted close to the working surfaces on two of the bearings.

Journal, Bearings, and Lubricant

The shaft used in these tests was made of a high-carbon, tungsten, tool steel heat-treated to a Brinell number of 179 and then ground and lapped. It was well-run-in as a result of use in previous running-in tests on similar bearings. The average diameter at the journals was 1.2504 in.

All three sets of bearings were made up of solid bronze sleeves with a lining of bearing metal about $^1/_{16}$ in. thick. They were faced off square to provide a bearing length of 1.250 in. and machined to size by the manufacturer.

The first set tested (bearing set 1) was lined with a high-lead babbitt. The reported mechanical properties (4, 5) of a similar alloy at 20 C were: Brinell number 22, impact strength (Izod test) 0.5 ft-lb, and stress to produce 0.3 per cent deformation in compression 5000 lb per sq in. The bronze sleeves were tinned in

[1] Publication approved by the Director of the National Bureau of Standards, Department of Commerce.

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[4] Numbers in parentheses refer to the Bibliography at the end of the paper.

Contributed by the Special Research Committee on Lubrication for presentation at the Annual Meeting of The American Society of Mechanical Engineers, to be held in New York, N. Y., December 6 to 10, 1937.

Discussion of this paper should be addressed to the Secretary, A.S.M.E., and will be accepted until January 10, 1938, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

a bath of pure tin at 620 F. The babbitt was cast by gravity at 900 F. The average of analyses of three samples taken after casting was 0.15 per cent copper, 3.28 per cent tin, 82.52 per cent lead, and 14.05 per cent antimony. The average diameter of the four bearings was 1.2514 in., making an average running clearance of 0.0010 in. and a clearance-diameter (C/D) ratio of about 0.0008.

The bearings in set 2 were lined with a high-tin babbitt. The reported mechanical properties of a similar alloy at 20 C were: Brinell number 19, impact strength 2.5 ft-lb, and stress to produce 0.3 per cent deformation in compression 6300 lb per sq in. The bronze sleeves were tinned in a bath of pure tin at 620 F. The babbitt was centrifugally cast at a temperature of 940 F and a speed of 1150 rpm. The average of analyses of two samples taken after casting was 4.50 per cent copper, 90.33 per cent tin, 0.40 per cent lead, and 4.77 per cent antimony. The average diameter of the set was 1.2515 in., making an average clearance of 0.0011 in. and a C/D ratio of about 0.0009.

The bearings of set 3 were lined with lead hardened with calcium and barium. The reported mechanical properties of a similar alloy at 20 C were: Brinell number 26, impact strength 1.3 ft-lb, and stress to produce 0.3 per cent deformation 7600 lb per sq in. These were centrifugally cast at 920 F and a speed of 1150 rpm. Pure tin at 640 F was used as a bonding material. Analyses of samples of this alloy after casting were not obtained. The nominal composition, however, was 97.5 per cent lead, 1.75 per cent barium, and 0.75 per cent calcium. The average diameter of the four bearings was 1.2516 in. making an average clearance of 0.0012 in. and a C/D ratio of about 0.0010.

The lubricant used in all the tests was a mineral spindle oil having absolute viscosities of 54 centipoises at 20 C, 42 centipoises at 25 C, and 33 centipoises at 30 C.

Test Results

The friction data obtained with the three sets of bearings at the various operating conditions are given in Tables 1, 2, and 3. Each test run, for a given set of bearings at a given degree of running in, was made in the decreasing order of ZN/P, using a constant load and changing the value of ZN/P by reducing the speed. These data are shown graphically by the f versus ZN/P curves in Figs. 2, 3, and 4.

(*a*) *General Effects of Running In.* The curves in Figs. 2, 3, and 4 are in substantial agreement with the findings in the previous investigation. They indicate that the major effect of running in upon the frictional characteristics of a journal bearing is to decrease the value of ZN/P at the point of minimum f as the amount of running in is increased. This characteristic is of significance from the standpoint of operation in that it tends to increase the factor of safety (1). There was an indication also that running in caused a slight increase in the slope of the f versus ZN/P curve at values of ZN/P just above the point of minimum f.

(*b*) *Differences in Running-In Characteristics.* A comparison of the running-in characteristics of the three sets of bearings is given in Fig. 5, where the frictional work done in running in is plotted against the values of ZN/P at minimum f. Included in this figure are the data obtained in the previous running-in tests, curve A representing high-tin babbitt bearings (C/D ratio 0.0022) and B high-tin babbitt bearings (C/D ratio 0.00045), and also data from another investigation (6) curve C representing leaded-bronze bearings (C/D ratio 0.0012).

From this figure it is seen that the bearings tend to fall into two groups, one, consisting of the leaded-bronze and hardened-lead bearings, being relatively hard to run in, and the other, consisting of the high-lead and high-tin babbitt bearings being relatively easy to run in. The figure indicates also that the points of mini-

TABLE 1 RUNNING-IN DATA, BEARING SET NO. 1

(High-lead babbitt; diameter $1^1/_4$ in., length, $1^1/_4$ in.)

Run No. 1, No running in, load 51.8 lb

Rpm	ZN/P	f
497	523.0	0.0344
360	380.0	0.0261
286	306.0	0.0218
223	239.0	0.0177
168	183.0	0.0140
122	135.0	0.0102
105	115.0	0.0087
84	94.0	0.0073
66	72.6	0.0060
46	50.6	0.0043
31	34.1	0.0043
24	26.4	0.0043
17	18.7	0.0048
14	14.9	0.0051

Run No. 2, 36 min running in, 107 ft-lb work, load 100.9 lb

Rpm	ZN/P	f
176	98.4	0.0074
108	60.4	0.0043
84	47.5	0.0034
69	38.7	0.0030
46	25.7	0.0026
36	20.1	0.0021
26	14.7	0.0022
20	11.3	0.0027
17	9.7	0.0040
12	7.0	0.0090

Run No. 3, 80 min running in, 212 ft-lb work, load 150.6 lb

Rpm	ZN/P	f
298	108.3	0.0077
208	75.8	0.0054
132	48.6	0.0037
120	44.4	0.0036
92	34.3	0.0030
67	25.0	0.0024
52	19.4	0.0021
39	14.5	0.0017
32	11.9	0.0017
27	9.9	0.0020
19	6.9	0.0026
14	5.3	0.0040

Run No. 4, 129 min running in, 328 ft-lb work, load 200.5 lb

Rpm	ZN/P	f
304	79.9	0.0059
198	52.4	0.0043
109	29.2	0.0027
82	22.0	0.0023
62	16.7	0.0019
45	12.1	0.0016
36	9.7	0.0014
32	8.7	0.0014
27	7.2	0.0014
21	5.6	0.0015
18	4.8	0.0018
14	3.8	0.0026

Run No. 5, 265 min running in, 632 ft-lb work, load 250.4 lb

Rpm	ZN/P	f
84	20.2	0.0021
57	13.6	0.0015
40	9.6	0.0013
32	7.7	0.0012
24	5.8	0.0011
18	4.3	0.0012
14	3.4	0.0015
8	1.9	0.0083

Run No. 6, 569 min running in, 973 ft-lb, work, load 300.3 lb

Rpm	ZN/P	f
82	15.2	0.0017
54	10.1	0.0013
36	6.6	0.0010
28	5.2	0.0010
19	3.6	0.0008
15	2.7	0.0011
13	2.3	0.0014
11	2.0	0.0018

Run No. 7 1017 min running in, 1726 ft-lb work, load 300.3 lb

Rpm	ZN/P	f
42	8.8	0.0012
30	6.1	0.0010
19	4.0	0.0009
14	2.8	0.0009
10	2.1	0.0011
8	1.6	0.0025

NOTE: If desired, the values of the viscosity in centipoises, Z, for each operating condition may be computed from the given values of ZN/P, speed, load, and bearing dimensions.

TABLE 2 RUNNING-IN DATA, BEARING SET NO. 2

(High-tin babbitt, diameter, $1^1/_4$ in., length $1^1/_4$ in.)

Run No. 1, No running in, load 100.9 lb

Rpm	ZN/P	f
362	173.0	0.0102
280	140.0	0.0083
94	48.0	0.0037
61	31.4	0.0028
46	23.6	0.0037
30	15.6	0.0042
17	8.6	0.0079

Run No. 2, 34 min running in, 167 ft-lb work, load 100.9 lb

Rpm	ZN/P	f
387	197.0	0.0113
290	150.0	0.0084
202	105.0	0.0062
99	53.0	0.0035
70	38.0	0.0028
51	27.4	0.0025
33	17.6	0.0019
24	12.7	0.0025

Run No. 3, 118 min running in, 329 ft-lb work, load 200.5 lb

Rpm	ZN/P	f
472	112.0	0.0064
306	74.5	0.0045
200	49.1	0.0033
101	26.3	0.0021
56	14.5	0.0016
44	11.5	0.0012
31	7.9	0.0013
24	6.2	0.0014
18	4.6	0.0025

Run No. 4, 285 min running in, 682 ft-lb work, load 300.3 lb

Rpm	ZN/P	f
359	60.0	0.0036
215	36.2	0.0026
103	17.4	0.0016
78	13.5	0.0011
53	9.0	0.0010
38	6.5	0.0010
27	4.6	0.0009
16	2.8	0.0039

Run No. 5, 353 min running in, 860 ft-lb work, load 300.3 lb

Rpm	ZN/P	f
80	13.7	0.0013
55	9.5	0.0012
43	7.5	0.0011
37	6.3	0.0010
29	4.9	0.0009
23	4.0	0.0009
17	3.0	0.0012
14	2.3	0.0033

Run No. 6 , 1059 min running in, 1677 ft-lb work, load 300.3

Rpm	ZN/P	f
47	9.6	0.0010
27	5.5	0.0009
15	3.1	0.0008
13	2.7	0.0007
10	2.1	0.0007
8	1.6	0.0009
7	1.4	0.0038

TABLE 3 RUNNING-IN DATA, BEARING SET NO. 3

(Hardened-lead alloy; diameter, $1^1/_4$ in.; length $1^1/_4$ in.)

Run No. 1, No running in, load 100.9 lb

Rpm	ZN/P	f
479	268.0	0.0154
364	220.0	0.0124
242	147.0	0.0092
161	101 0	0.0076
94	61.0	0.0077
101	67.0	0.0079
73	48.4	0.0077
58	38.6	0.0084
48	31.5	0.0100
36	23.9	0.0121

Run No. 2, 153 min running in, 2618 ft-lb work, load 100.9 lb

Rpm	ZN/P	f
518	294.0	0.0163
316	182.0	0.0111
192	115.0	0.0081
102	62.0	0.0063
57	36.0	0.0064
41	25.9	0.0082
27	17.0	0.0119

Run No. 3, 257 min running in, 4279 ft-lb work, load 200.5 lb

Rpm	ZN/P	f
536	154.0	0.0095
374	110.0	0.0073
217	67.0	0.0052
170	53.5	0.0046
100	32.2	0.0041
71	22.9	0.0040
51	16.1	0.0047
33	10.4	0.0068

Run No. 4, 432 min running in, 7361 ft-lb work, load 200.5 lb

Rpm	ZN/P	f
528	147.0	0.0088
318	92.0	0.0060
174	52.5	0.0042
102	31.0	0.0036
65	20.4	0.0036
48	14.9	0.0042
36	11.3	0.0049
29	9.1	0.0066

Run No. 5, 946 min running in, 18,942 ft-lb work, load 300.3 lb

Rpm	ZN/P	f
568	99.0	0.0059
319	57.0	0.0038
212	39.3	0.0031
105	20.0	0.0024
65	12.5	0.0023
46	8.9	0.0025
32	6.2	0.0037
24	4.6	0.0078

Run No. 6, 1465 min running in, 27,829 ft-lb work, load 300.3 lb

Rpm	ZN/P	f
546	94.0	0.0055
334	58.0	0.0039
198	36.5	0.0028
104	19.6	0.0020
52	10.0	0.0018
41	7.9	0.0017
30	5.9	0.0018
22	4.3	0.0033

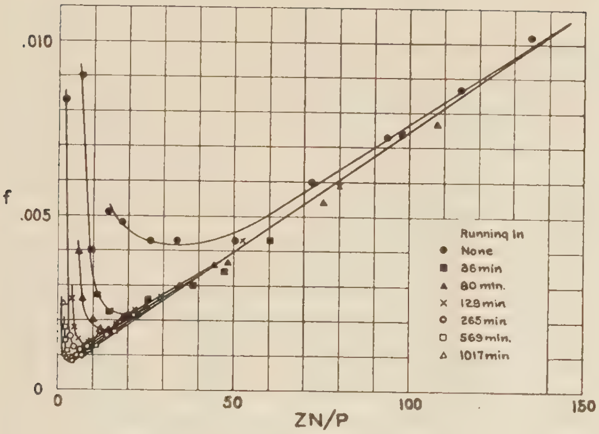

FIG. 2 JOURNAL FRICTION CURVES, SHOWING PROGRESSIVE AMOUNTS OF RUNNING IN

(Set 1: high-lead babbitt bearings.)

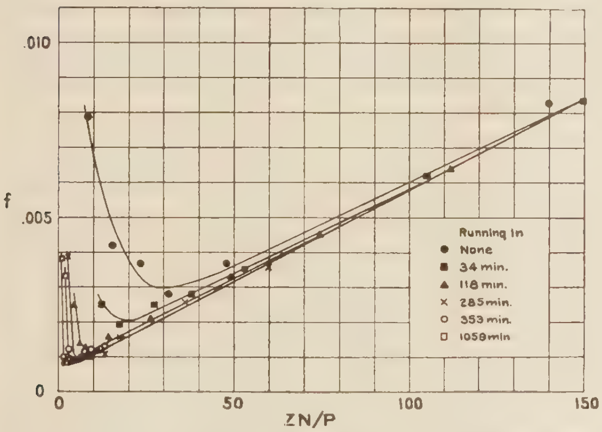

FIG. 3 JOURNAL FRICTION CURVES, SHOWING PROGRESSIVE AMOUNTS OF RUNNING IN

(Set 2: high-tin babbitt bearings.)

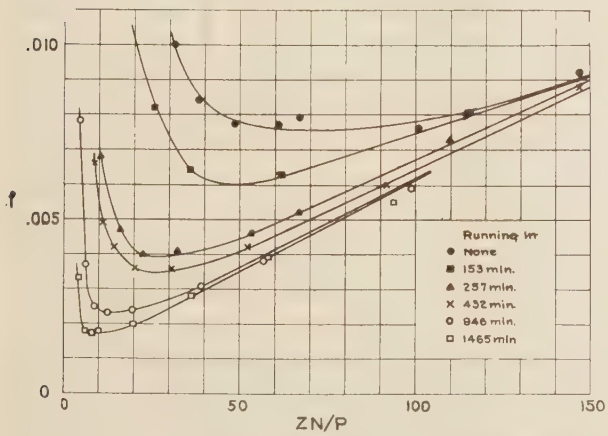

FIG. 4 JOURNAL FRICTION CURVES, SHOWING PROGRESSIVE AMOUNTS OF RUNNING IN

(Set 3: hardened-lead bearings.)

mum f for comparable conditions[5] of running in occur at higher values of ZN/P for the bronze and hardened-lead bearings than for the babbitt bearings.

[5] Because of the scale to which Fig. 5 is drawn the upper portions of curves 3 and C were omitted.

The data are too limited to indicate whether or not there is a general relation between the mechanical properties and the running-in characteristics of bearing metals. The higher values for the hardness and compressive strength of the hardened-lead alloy used in set 3 possibly are in agreement with the results of the running-in tests which showed that these bearings were harder to run in than the high-lead or high-tin babbitt bearings. On the other hand, from the mechanical properties of the metals used in sets 1 and 2 it might not be expected that these bearings would be so nearly alike in running-in characteristics as the tests indicate. Relatively large deposits of worn-off particles of bearing metal were found in the clearance spaces after the tests on set 3 were completed. These deposits possibly account, at least in part, for the greater difficulty in running in, for the higher values of ZN/P at minimum f, and also for the generally higher values of minimum f.

It is difficult to suggest the causes of the differences shown by the curves 2, A, and B since these differences are probably the result of a combination of factors. All three sets had different clearances but the results do not follow the order of change in clearance. There was some difference in the metals, the nominal composition of the babbitt used in sets A and B being 85 per cent tin, 7.5 per cent copper, and 7.5 per cent antimony. This

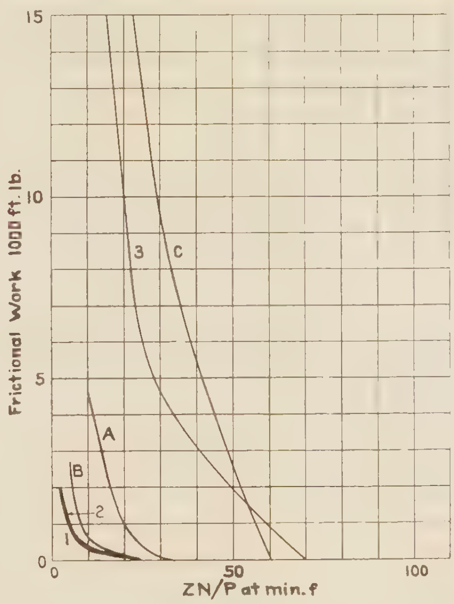

FIG. 5 RELATION BETWEEN LOCATION OF THE MINIMUM POINT AND THE FRICTIONAL WORK DONE ON THE VARIOUS BEARINGS

difference in composition may account for the better running-in characteristics shown by set 2. Another possible reason is that the journals may have been improved by a number of tests that had been run between the tests on set B and set 2. It should be noted also that in plotting the data as in Fig. 5 no provision is made for the possibility that the effects of a given amount of frictional work may be different at different loads and speeds, and the experience gained in the earlier tests tended toward an improved technique of running in for the later tests. Other factors that may be significant are geometrical variations and differences in methods of casting and finishing the bearing metals.

(c) *Effect of Changes in Load.* In another investigation (6) it was found that for values of ZN/P near the point of minimum f, changes in load had marked effects upon the frictional characteristics of a well-run-in journal bearing. An increase in load

tended to decrease the value of ZN/P at minimum f and analysis of the data indicated that in this "thin-film" region the frictional characteristics of a bearing could be represented with reasonable accuracy by a single curve if f was plotted against $ZN/\sqrt{P}$.

The final test runs for all the bearings were made at a load of 300 lb, thus the curves representing these runs in Figs. 2, 3, and 4 provide a direct comparison of the frictional characteristics of the three sets of bearings in the well-run-in condition.

The initial runs, however were not all made at the same load and in all cases were made at loads lower than 300 lb in order to minimize the risk of scoring the bearings. It is probable therefore, that the corresponding points on the curves in Fig. 5 do not provide a direct comparison of the relative locations of the points of minimum f for all the bearings at the initial conditions. This may be obtained by correcting the observed values of ZN/P at minimum f to conform to a load of 300 lb, basing the correction factors upon the findings of the other investigation. The loads used in the initial runs with sets 1, 2, 3, A, B, and C were 50, 100, 100, 200, 50, and 150 lb, respectively, and the corrected values of ZN/P at minimum f for the respective bearings are 14, 17, 41, 26, 11, and 43. The data for the initial runs are of such a nature as to make it difficult to estimate with accuracy the location of the point of minimum f, thus the corrected values given are approximations only. While it is probably not significant, it is of interest that set B has the lowest corrected value. The bearings in this set were finished with a special reamer and had the appearance of being smoother than the other bearings.

Conclusion

The characteristics of journal bearings indicated by the results of this and the previous investigations are summarized as follows:

(*a*) The major effect of running in upon the frictional characteristics of a journal bearing is to reduce the value of ZN/P at the point of minimum f and hence to tend to increase the factor of sofety.

(*b*) The amount of running in required to produce a given change in the location of the point of minimum f was greater for the leaded-bronze and the hardened-lead bearings than for the high-lead or high-tin babbitt bearings.

(*c*) In the region of thin-film lubrication under comparable conditions of operation and degree of running in the bronze and hardened-lead bearings had a higher friction than the babbitt bearings.

(*d*) The high-lead and high-tin babbitt bearings that were essentially alike in clearance and finish were also practically alike in running-in characteristics.

(*e*) The data are too limited to indicate whether or not there is a general relation between the mechanical properties and the running-in characteristics of bearing metals.

While these tests provide a general indication as to the running-in and frictional characteristics of the various bearing metals used, they are not of sufficient scope to indicate the effect of all the factors involved. Systematic investigations of the effects of changes in length, diameter, clearance, bearing metal, technique in casting and finishing the metal, and method of running in, are suggested as promising fields of study. At the present time the need for such investigations is increasing to a marked extent because of the use of new types of bearing metals. Also there is a growing tendency toward the use of oils that are compounded for the purpose of increasing either the oiliness or the so-called film strength and there is a lack of information as to how these various compounds actually affect bearing performance.

Acknowledgment is made to the Bohn Aluminum & Brass Corporation for furnishing the bearings used in these tests.

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Investigation of the Oxidation of Metals by High-Temperature Steam[1]

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Inasmuch as steam temperatures in modern central stations are approaching those used for the commercial production of hydrogen by reaction between steam and iron, an investigation was undertaken at Purdue University of the oxidation by steam of the various steels that are available for high-temperature steam service. Apparatus was constructed and techniques developed for measuring the amount of oxidation due to temperatures up to 1200 F and pressures up to at least 1600 lb gage. Data are presented showing the effect of temperature from 800 to 1200 F on the oxidation of low-carbon steel in contact with steam at 1200 lb gage. The rate of oxidation of low-carbon steel at 1100 F is apparently the same at 400 and 1200 lb steam pressure. Apparatus was developed for operating seven tubes simultaneously under identical conditions and data are presented to show the comparative oxidation of six steels of different analyses. The investigation is being continued.

HYDROGEN is produced commercially in large quantities by passing steam through a bed of porous iron ore, which is maintained at a temperature of from 1000 F to 1500 F. The process may be represented by the following reaction

$$3Fe + 4H_2O = Fe_3O_4 + 4H_2 \qquad [1]$$

After about ten minutes of steaming during which approximately one third of the steam is converted into hydrogen, water gas is passed through the ore for twenty minutes to reduce the magnetic iron oxide, Fe_3O_4, to iron or a lower oxide of iron. The cycle is then repeated. Alloys of iron, especially manganese, chromium, tungsten, and titanium, have shown higher rates of reactivity than the cheaper iron ores, due to the lesser tendency of alloys of iron to sinter with local overheating. A working temperature of 650 C or about 1200 F is common.

Most of the central-station installations now under construction in the high-pressure range, about 1400 lb steam pressure, are designed to operate on steam at 900 to 925 F. Better efficiency would be obtained by using still higher steam temperatures. Inasmuch as the inner surface of a superheater tube, subject to heat transfer, is at a higher temperature than the steam in the tube, and this temperature difference may be considerable during starting or due to poor steam distribution, it is apparent that superheater tubes may be operated at temperatures which approach those used for the commercial production of hydrogen from steam. In view of the reduced fuel consumption which would be possible by increasing the initial steam temperature to between 1000 and 1200 F, a knowledge of the extent to which steam reacts with alloy steels at these temperatures is of considerable value in determining the best material for service under such high-temperature conditions. The hot end of the high-temperature superheater is now made of alloy steel but few data are available as to the effect of steam on the various alloy steels that have been proposed for this service. The investigation now being carried on at Purdue University has for its objective, therefore, the determination of the extent of oxidation by high-temperature steam of the various steels that are available for service at the temperatures which are being used at present or may be used in the future in modern steam-power stations.

Test Specimens Proposed

The following materials have been suggested as being suitable for inclusion in the test program:

(1) Plain carbon steel
(2) Plain carbon steel, calorized
(3) Carbon; 0.5 per cent Mo
(4) Carbon-molybdenum steel, calorized
(5) Carbon; 1 per cent Mo
(6) Croloy 2: 2 per cent Cr; 0.5 Mo
(7) Croloy 3: 3 per cent Cr; 0.8 per cent Mo
(8) Timken DM: 1.25 per cent Cr; 0.55 per cent Mo
(9) Silicon-molybdenum: 1.5 per cent Si; 0.5 per cent Mo
(10) Timken Sicromo: 2.46 per cent Cr; 0.50 per cent Mo; 0.94 per cent Si
(11) 4–6 per cent Cr; 0.5 per cent Mo
(12) 4–6 per cent Cr; 0.5 per cent Mo; titanium equal to 4 times carbon
(13) 4–6 per cent Cr; 0.5 per cent Mo; columbium equal to 8 times carbon
(14) 9 per cent Cr; 1.5 per cent Mo
(15) KA2S: 18 per cent Cr; 8 per cent Ni
(16) KA2B which is same as KA2S with 2.5 per cent Si
(17) 18 per cent Cr; 8 per cent Ni; titanium equal to 4 times carbon

[1] Progress report of the A.S.M.E. Special Research Committee on Critical-Pressure Steam Boilers.

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1938, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

(18) 18 per cent Cr; 8 per cent Ni; columbium equal to 8 times carbon
(19) S.A.E. No. 6120: 1 per cent Cr; 0.18 per cent Va
(20) USS Cor-Ten: 1.12 per cent Cr; 0.5 per cent Cu
(21) Bain alloy: 1.88 per cent Cr; 0.6 per cent Mo
(22) R R 11 alloy: 12 per cent Cr; 0.5 per cent Ni
(23) 19 per cent Cr; 9 per cent Ni; 0.05 per cent C

Low-carbon steel being presumably the most susceptible to oxidation in the presence of steam of any of the materials in the list and, therefore, the most affected by changes in the variables which enter into the problem, it was decided to adopt low-carbon steel as the basic material with which to compare the behavior of the various steels in the foregoing list. One hundred linear feet of $1^1/_2$-in. OD × 1-in. ID low-carbon seamless steel tubing from the same heat were supplied for this part of the investigation through the courtesy of the Babock & Wilcox Company. Apparatus was designed and constructed to permit the determination of the influence of such variables as pressure, temperature, and steam velocity upon the reaction between steam and low-carbon-steel tubing. It was planned to extend the investigation to the various alloy steels with such modifications in equipment and procedure as would be indicated during the study of the behavior of low-carbon steel in contact with steam at high temperatures.

Description of Apparatus

Inasmuch as the reaction between steam and metals is presumably a surface phenomenon which occurs at the temperature of the metal surface, it was decided to superheat the steam to the desired temperature and then pass it through a uniformly heated test specimen which was heated electrically to the same temperature as the steam. The change in the gas content and composition of the steam flowing through the test tube was assumed to be a measure of the reaction in the tube. This was to be determined by condensing all of the steam passing through the test section as well as a sample of the steam at the entrance to the test section, separating the dissolved gases from the condensate and collecting and analyzing them in a Burrell apparatus. For most accurate results, it is obvious that the gas content of the steam at the entrance to the test section should be kept to a minimum.

In 1927, C. H. Fellows and M. G. Amick conducted experiments at The Detroit Edison Company on "The Trend of the Reaction Between Steam and Metals at Elevated Temperatures," using low-carbon-steel and Enduro (16.35 per cent Cr) tubes. The experimental setup was essentially the same as the one outlined in the preceding paragraph except that steam was introduced into the test section at about 750 F and also at saturation temperature, 420 lb steam pressure, and was heated during its passage through the test section, thus causing uncertainty as to the actual surface temperature at which the reaction was taking place.

To generate steam in the Purdue University laboratory at high pressures and temperatures, a small gas-fired once-through steam generator was constructed by coiling about fifty linear feet of $^3/_8$-in. extra-heavy iron-pipe-size seamless low-carbon-steel tubing and thirty linear feet of $^3/_8$-in. extra-heavy iron-pipe-size 18-8 stainless-steel tubing stabilized with columbium. The stainless-steel tubing was used for the superheating section.

While it was possible to control, within close limits, the steam pressure and steam temperature at the outlet of this steam generator, the steam at 1200 F was found to contain an excessive quantity of gas, principally hydrogen. This trouble was thought to be due to unequal heating of the tubing, resulting in localized hot spots. Efforts to overcome this situation by rearrangement of the heating surface were unsuccessful. It was decided, therefore, to build a small water-tube boiler and to complete the superheating of the steam in an electrically heated tube to eliminate the possibility of hot spots in the high-temperature end of the superheater.

Fig. 1 shows diagrammatically the arrangement of the apparatus as finally constructed. The water-tube boiler has an evaporative capacitive of about 50 lb of water per hour. The drum was machined from a scrap piece of 10-in. locomotive axle and the boiler tubes were made of 1-in. extra-heavy iron-pipe-size seamless-steel tubing. The primary superheater is the coil of 18-8 stainless-steel tubing stabilized with columbium which was used in the once-through steam generator previously mentioned. At 1200 lb operating pressure, the steam leaves the primary superheater at 800 F. The final superheating to the test temperature is performed in 20 linear feet of $1^1/_2$-in. OD × $1^1/_{16}$-in. ID 18-8 stainless-steel stabilized with columbium and containing a core of unstabilized 18-8 which is 1-in. OD. Chromel A resistance wire, insulated with porcelain beads, is wound around the tubing and is arranged in four sections which may be connected in series or in parallel with suitable control resistances. Superheated steam from the secondary superheater may be throttled, condensed in a coil of copper tubing placed in a can through which cold water flows, and weighed, or it may flow through two stop valves in series with a vent valve between them to the test section. The test section consists of a 3-ft length of $1^1/_2$-in. OD × 1-in. ID tubing of the material being studied, partially closed at each end by a screwed reducer and heated uniformly by an electric heater wound on a

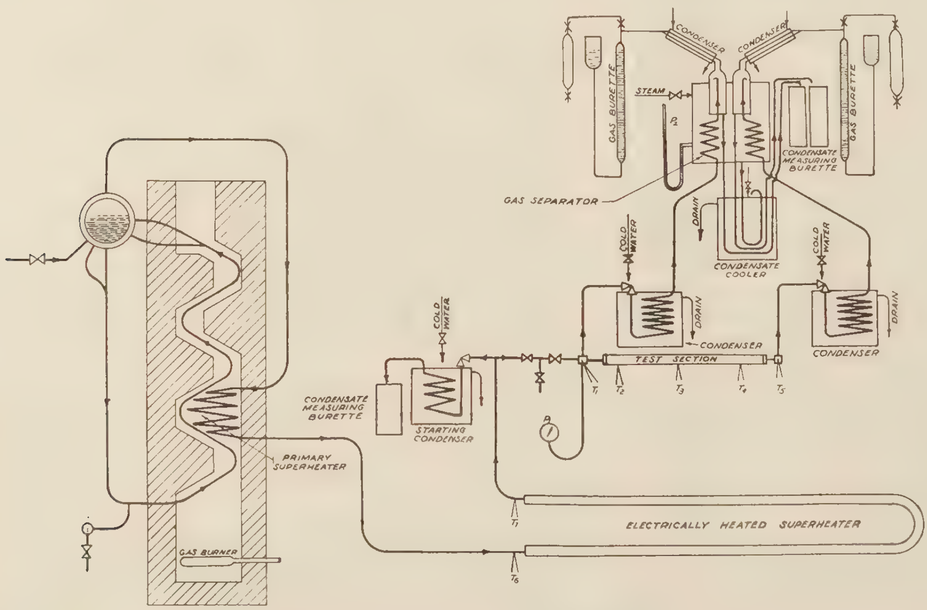

Fig. 1 Diagrammatic Arrangement of High-Pressure Apparatus

piece of $1^1/_2$-in. standard pipe which will slip over the test section. Immediately ahead of the test section is a fitting into which are connected a pressure gage, thermocouple, and steam sampling line. Another fitting on the discharge side of the test section contains a second thermocouple. Each thermocouple is a small well of 18-8 stainless steel threaded to screw into a $^1/_4$-in. pipe thread, with chromel and alumel thermocouple wires welded into the bottom of the well in such a way that the weld is immersed in the steam. Three thermocouples are peened to the outside surface of the test section as indicated in Fig. 1 and the electrical input to the heater is regulated so that the five thermocouples indicate the same temperature. Then the heater supplies the radiation loss, and the steam and internal pipe surface are at the same temperature. Thus the temperature at which the reaction takes place is known.

About one half of the steam passing the superheater is removed through the sampling line ahead of the test section, throttled, and condensed in a copper tube immersed in cold water, while the rest of the steam passes through the test section and is in turn throttled and condensed. The condenser cooling water is discharged onto the throttle valves to keep them cool, improve the regulation, and reduce the steam temperature before reducing the pressure.

The condensate from the two condensers flows upward to the gas separator, the details of which are shown in Fig. 2. It consists of a section of 8-in. steel pipe closed at the lower end by a blind flange and with a welded cover plate on the upper end into which are welded two 2-in. nipples that are closed on their lower ends by welding. The condensate enters at A and B, passes upward through the copper heating coils S to the glass bells C which are set in rubber stoppers in the upper ends of the 2-in. nipples. Steam is supplied to the gas separator and the pressure, measured by a manometer, is held constant at 25 in. of water which causes violent boiling of the condensate in the glass bells. The condensate is drained through the copper lines D and cooler E to the measuring burettes. The elevation of the ends of the condensate-discharge lines is adjusted to maintain a water level in the glass separating bells. The gases which are liberated from the boiling condensate pass through the auxiliary condensers O to the gas-measuring burettes where they are collected and measured at atmospheric pressure. The gas burettes and water bottles contain a saturated salt solution colored with methyl orange and sulphuric acid and saturated with commercial hydrogen. The gases are stored over mercury and analyzed in a standard Burrell apparatus.

All piping, fittings, valves, superheaters, thermocouple bodies, and all other metal in contact with steam between the outlet of the boiler drum and the condenser coils on the discharge side of the water-cooled throttle valves are made of 18-8 stainless steel, with the single exception of the test specimen. Most of the joints were originally screwed connections, using standard $^1/_4$- or $^3/_8$-in. pipe threads and a good high-temperature thread lubricant. Most of these screwed connections were finally welded to reduce leakage difficulties.

Make-up water is obtained by condensing steam from the main in the steam laboratory of Purdue University in a coil of seamless copper tubing having a condensing capacity of about 125 lb per hr. The hot-well capacity is about 500 lb and contains a steam coil in which 50 lb steam pressure is maintained during tests. About 80 lb of water is evaporated per hour in the hot well and vented to the atmosphere to remove dissolved gases. The remainder of the output of the make-up condenser is discharged from the hot well to the sewer so as to prevent dissolved iron or other impurities from concentrating in the hot-well water. A $1^1/_2 \times 8$-in. vertical triplex plunger pump is used for boiler-feed purposes and the excess water from the pump is delivered back

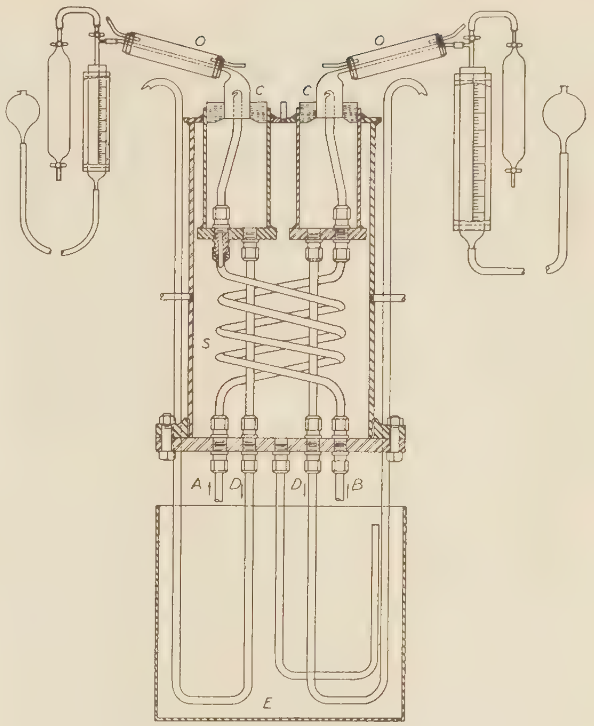

Fig. 2 Details of Gas Separator

to the hot well through a by-pass valve. A pH value of 10 to 11 is maintained in the boiler water by the use of sodium hydroxide.

Fifteen grams of sodium hydroxide is required to condition the water used to fill an empty boiler. Mallinkrodt technical reagent, which contains 1.75 per cent of Na_2CO_3, is used for this purpose. The gas separators are so sensitive that it is necessary to operate the boiler for a day following the introduction of the sodium hydroxide into the boiler to void the CO_2 liberated by the breakdown of this small amount of Na_2CO_3.

Prior to conducting a test, the test specimen is sandblasted to remove all scale and to bring the internal surface to a uniform condition, after which it is filled with nitrogen and the ends are corked and sealed with paraffin. After a test specimen is inserted, the thermocouples, heater, and insulation are installed; the system is filled with nitrogen; and the temperature is raised to the desired operating temperature, as indicated by the three thermocouples on the surface of the test specimen. Meanwhile, the steam generator and secondary superheater are brought up to normal operating conditions, and the steam is discharged through the starting condenser. After the flow rate and steam temperatures are stabilized, the stop valves to the test section are opened while the starting condenser is shut off, and the flow through the two condensers is adjusted by the throttle valves. Thirty minutes after the steam first enters the test section, the collection of gas is started.

Experiences With 18-8 and 25-20 Superheater and Piping Material

When the equipment was first placed in operation, excessive quantities of gas appeared in the steam at the end of the electrically heated superheater element at steam temperatures above 1000 F. As the authors had been advised that steam should react with 18-8 steel to only a limited extent at temperatures below 1200 F, considerable time was spent in checking into the reason for the excessive gas evolution. The gas evolution ceased

at steam temperatures below 975 F. It also apparently ceased if the steam pressure in the gas separator was reduced and appeared again when the pressure was returned to normal, thus indicating the necessity of employing a well-designed gas separator in which the condensate is boiled to detect the presence of the gas. The boiler feed pump packing and make-up and feedwater were checked for organic material. Oil deflectors were placed below the feed-pump crossheads to prevent lubricating oil from working down the pump plungers. Oil-deflecting shields were placed over the pump cylinder block to prevent oil from falling onto the block and working through the pump packing. Several pump packings were tried and Carbonite packing, made from asbestos, graphite, and a small amount of easily voided binder was selected. The packing was set up loose enough to allow some leakage of water to reduce entrainment of air and impurities by the descending pump plungers. None of these changes in the equipment or methods of operation eliminated the gas which appeared as soon as the temperature of the steam from the electrically heated 18-8 superheater exceeded 975 F.

A section of $^5/_8$-in. OD $\times$ $^3/_8$-in. ID, 25 per cent chromium, 20 per cent nickel tubing was then connected in series with the 18-8 superheater with a steam-sampling connection at the end of each element. The 25-20 tube was also heated electrically to insure uniform heating. On the first test, with steam entering the 25-20 tube at 950 F and leaving at 1030 F, 550 cu cm of gas, which analyzed 93 per cent hydrogen, was collected in 33 min. The rate of gas evolution decreased with increase of time.

Runs were made on two successive days with the two superheater materials in series and with steam at 1200 lb and 1100 F leaving each tube. The gas evolved, expressed in cubic centimeters per hour, from a steam flow of 150 cu cm of condensate per minute was as follows:

	18-8 Tube	25-20 Tube
First run	104	76
Second run	88	52

The gas samples analyzed 80–88 per cent H_2, about three per cent CO_2, and the rest nitrogen. It is probable that the entire 15 linear feet of 25-20 tube was at 1100 F, while the amount of surface in the 18-8 tube at a temperature above 1000 F is unknown since the steam was being superheated in this element.

Experience with these materials has led to the following conclusions:

(1) Steam at 1200 lb and 1000 F will attack 18-8 and 25-20 stainless steels with the evolution of hydrogen and the formation of a black layer of oxide which decreases the rate of reaction as the time element increases.

(2) In view of the reaction between steam and these high-chromium stainless steels, it is probable that any commercially available superheater-material will be subject to a certain amount of oxidation by high-pressure high-temperature steam.

(3) It is probable that for extrahigh steam temperatures, alloy materials should be used which will react with the steam to produce a thin, dense, tightly adherent oxide layer that offers maximum resistance to the penetration of the oxidizing agent and will not crack or spall when subjected to rapid or extreme changes in temperature.

It should be noted that the black-oxide layer formed on stainless steel in contact with steam at 1200 lb and 1000 F is entirely different, at least in appearance, from the microscopically thin, bright, oxide layer that is formed in air at room temperature.

Gas Evolution From Low-Carbon Steel

To check the accuracy with which results could be reproduced a number of test runs were made on specimens of low-carbon steel tubing 3 ft in length and $1^1/_2$-in. OD $\times$ 1-in. ID, with steam at 1200 lb pressure and 1000 F. The total amount of gas collected from the test specimen and the amount of hydrogen in this gas showed variations of as much as 2 to 1 between tests which were apparently identical. Steam and tube temperatures, pressures, and flow rates could be held to within $\pm$ 1 per cent of the desired value without difficulty. Condenser-cooling-water temperature, gas-separator steam pressure, hot-well-heater pressure, pump speed, boiler-water conditions, and starting operations were examined carefully and standardized. Inasmuch as erratic results continued, suspicion was directed to the test tube itself.

It was known that gases are occluded in steel and that they cause difficulties in high-vacuum work. It was suspected that they were responsible for some of the troubles encountered at high temperatures. Five different tubes of low-carbon steel $1^1/_2$-in. OD $\times$ 1-in. ID and three ft long, which had been sandblasted internally and filled with nitrogen at atmospheric pressure and sealed, were mounted in turn in an electric heater, connected to a tank of commercial nitrogen at one end, and provided with a small-diameter outlet at the other. Nitrogen was allowed to flow through the tube to remove all air. The nitrogen tank was disconnected after the inlet line had been closed by three valves in series. The tube was heated to 1000 F and after steady temperature had been established, a gas-collecting burette and leveling bottle were used to collect the gases being discharged from the tube. Atmospheric pressure was maintained in the tube by the leveling bottle and no gas was collected until the temperatures had stabilized at a constant value. Tubes *A*, *B*, *C*, and *E*, Fig. 3 and Table 1, were maintained at a temperature at 1000 F. The inner surface of the tubes showed evidence of the formation of a black layer, possibly of oxidized material. Therefore, tube *D* was heated only to 750 F to avoid changing the visible appearance of the internal tube surface. Fifty-eight cubic centimeters of gas were collected from tube *A* (*A*-1 in Fig. 3) after which it was allowed to cool overnight while full of nitrogen at atmospheric pressure. On the second day, *A*-2 Fig. 3, 285 cu cm of gas were collected from the same tube after the temperature had been stabilized at 1000 F. An

TABLE 1 ANALYSES OF GASES LIBERATED FROM LOW-CARBON STEEL TUBES CONTAINING NITROGEN AT ATMOSPHERIC PRESSURE

Tube no.	*A*-1	*A*-2	*B*	*C*	*D*	*E*
Temperature, F	1000	1000	1000	1000	750	1000
Date	5/11/37	5/12/37	5/13/37	5/15/37	5/18/37	5/20/37
Gas collected, cu cm	58	285	285	40	94	168
Analysis, per cent						
CO_2	6.55	4.33	5.27	....	7.80	3.15
O_2	2.29	0.82	0.71	....	1.28	1.59
CO	0.86	0.0	5.17	....	0.61	2.99
H_2	53.1	17.8	21.85	....	10.60	5.19
N_2	37.2	77.5	67.0	....	79.71	87.08
H_2 collected, cu cm	31	51	62	....	10	9
Time required to reach equilibrium, hr	1	..	7	6	15	10

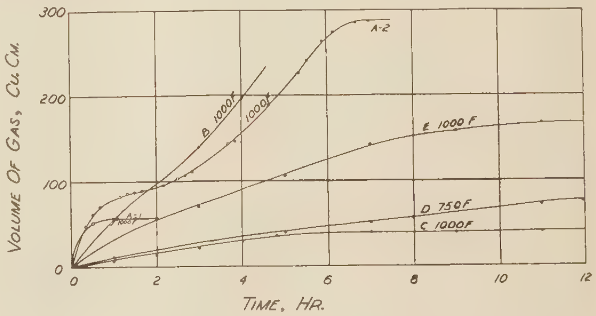

Fig. 3 Gas Evolution From Low-Carbon Steel Tubes
(These tubes were $1^1/_2$ in. OD, 1 in. ID, and 36 in. long and were filled with nitrogen.)

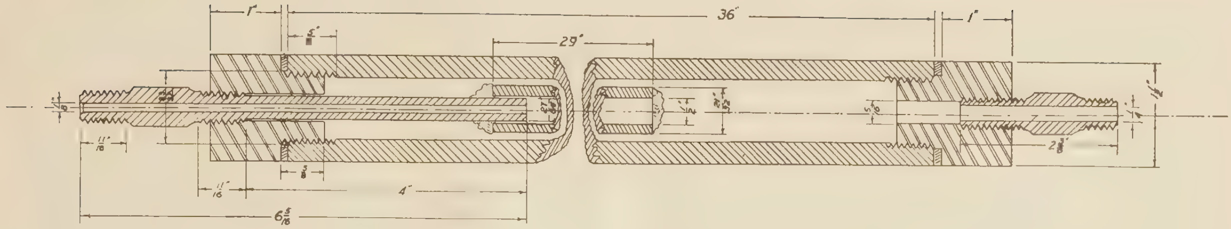

Fig. 4 Detail of Apparatus Used for Determining the Diffusion Rate of Gases Through Steel at High Temperatures

insufficient amount of gas was collected from tube *C* to make possible an accurate analysis. The gas was expelled from the tube more quickly at 1000 F than at 750 F. A marked variation in results was obtained with different tubes at the same temperature and with the same tube at the same temperature on successive days as shown in Fig. 3 and Table 1. The erratic nature of these results furnished a possible explanation for the inability to check steam-metal reaction rates on the basis of hydrogen evolution from tubes on successive tests where all variables were presumably under control and eliminated. It also caused speculation as to the amount of gas occluded in the steel, the manner in which it is released on heating, the extent to which gas is absorbed on cooling from a high temperature, and the extent to which it will diffuse through a layer of steel at high temperatures.

The apparatus shown in Fig. 4 was constructed to make some comparative measurements of the rate of diffusion of gases through steel at high temperatures. A 30-in. piece of $^3/_8$-in. standard iron-pipe size low-carbon seamless steel tubing was sealed at both ends by welding and a nipple was welded into one end. A lathe cut was taken over the external surface to remove all paint, oil, and mill scale. This tube was inserted into a 3-ft section of $1^1/_2$-in. OD $\times$ 1-in. ID tubing closed at both ends by plugs. Gas could be introduced into the annular space between the tubes from one end, while gas could be introduced into or removed from the inner tube at the other end of the apparatus. An electric heater was placed over the outer tube and suitable peened thermocouples made possible the adjustment of the power input so as to attain any desired tube temperature up to 1200 F. The inner tube and nipple were tested for tightness by immersion in water when filled with nitrogen at 600 lb per sq in.

With the tube at 1000 F, hydrogen maintained in the annular space at a constant pressure of 100 lb per sq in. gage and nitrogen in the inner tube at atmospheric pressure at the start of the test, 3200 cu cm of practically pure hydrogen were collected from the inner tube in a period of 19 hr of steady-temperature operation. Fig. 5, plotted from the data, indicates a uniform rate of diffusion of the hydrogen through the inner tube with respect to time. At the conclusion of this test, the temperature was decreased to 800 F and the annular space was filled with oxygen at 100 lb gage pressure. No gas could be collected from the inner tube, thus indicating not only the absence of oxygen diffusion through the metal but demonstrating the tightness of the system to ordinary leakage.

To determine the effect of differential pressure and temperature upon the rate of diffusion of hydrogen through a steel tube, a second apparatus was constructed in accordance with Fig. 4. Hydrogen was maintained in the annular space at a constant pressure of 100 lb per sq in. gage and the inner tube was filled at the start with hydrogen at atmospheric pressure. Gas was collected from the inner tube at atmospheric pressure. This constant differential pressure of 100 lb per sq in. was maintained across the inner tube while the apparatus was operated at several different temperatures within the range from 400 F to 1200 F to measure the effect of temperature upon the rate of hydrogen diffusion. The results shown in Fig. 6 indicate no appreciable diffusion below 600 F and a rate of diffusion which increases rapidly with temperatures above 600 F. The apparatus was also operated at a constant temperature of 1000 F with atmospheric pressure in the inner tube and hydrogen at pressures from 20 to 100 lb per sq in. gage around the tube. As shown by the plotted results of Fig. 6, the rate of hydrogen diffusion depends directly on the pressure difference across the metal when the temperature is constant. The annular space between the tubes was next filled with nitrogen at a constant pressure of 100 lb per sq in. gage and operated at 1000 F and also at 1200 F. No gas could be collected from the inner tube, thus demonstrating the tightness of the apparatus to ordinary leakage and the fact that nitrogen will not diffuse through the tube at temperatures up to 1200 F.

The results of these gas-diffusion tests indicate that while hydrogen will pass through low-carbon steel and nitrogen and

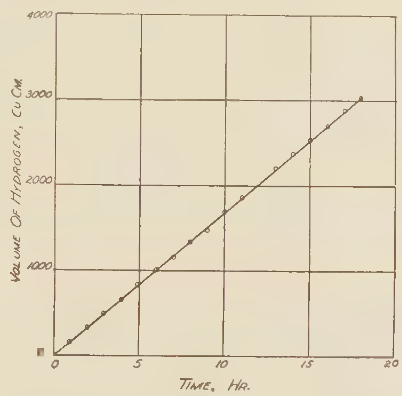

Fig. 5 Volume of Hydrogen Passing Through a Steel Tube at 1000 F

(This tube was 21/32 in. OD, $^1/_2$ in. ID, and 29 in. long. The pressure within the tube was atmospheric and that on the outside was 100 lb per sq in.)

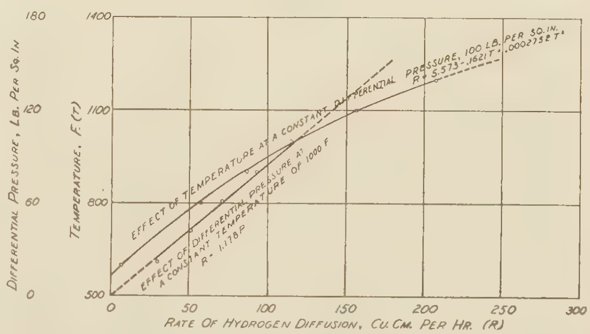

Fig. 6 Hydrogen Diffusion Through a Low-Carbon-Steel Tube

(This tube was $^{21}/_{32}$ in. OD, $^1/_2$ in. ID, and 29 in. long.)

oxygen will not do so within the temperature range of the steam-corrosion experiments, the amount of hydrogen passing through the tube is probably small at the low partial-hydrogen pressure existing within the test section. It is, therefore, doubtful if the inability to check the results of successive steam-corrosion tests under presumably identical conditions on the basis of hydrogen generated in the test section can be attributed to hydrogen diffusion through the test specimen.

To minimize the influence of occluded gases in the steel, as shown in Fig. 3, the starting cycle was modified. After the test specimen was inserted into the equipment, it was filled with nitrogen and heated electrically to 800 F for a period of 12 hr after which it was brought to operating temperature, the steam was admitted, the collection of gas was started 30 min later, and the gas collected in the first two hours was analyzed and then discarded. It was still impossible to check the results of successive duplicate tests on the basis of the hydrogen evolved in the test section. It was, accordingly, decided to base the results on the weight of oxide that could be removed from the internal surface of the test tube at the conclusion of each test.

Determination of Corrosion Products by Weight

The weight of scale was determined on a Becker balance capable of weighing a 100-g specimen with an accuracy of 1 part in 1,000,000. Within five minutes after the steam was shut off from the test section, it was removed from the apparatus, filled with nitrogen, and corked. Failure to carry out this procedure resulted in a red deposit forming in the tube. The tube was cut into six 4-in. lengths and machined carefully on the outside to remove the surplus steel to a shell thickness of about $^1/_{16}$ in., giving a specimen weight of about 100 g. Great care was necessary in making the final external cut as well as the end cuts to prevent chipping of the brittle scale. Each machined piece was wrapped in paper and handled with clean hands. On the recommendation of C. H. Fellows of The Detroit Edison Company, the following solvent and inhibitor was used to remove the scale with minimum attack on the parent metal:

Constituent	Parts by weight
Concentrated hydrochloric acid	100
Antimony oxide (Sb_2O_3)	2
Stannous chloride ($SnCl_2$)	5

After the specimen with its scale had been weighed, it was immersed in this solution, which was kept below room temperature, for a period of from 10 to 20 min. After removal from the solution, the specimen was washed in alcohol, dried, and again weighed. The length of specimen was measured by a 4-in. micrometer and the internal diameter was also measured after removal of the scale. Then the weight of scale was computed in terms of grams per square inch of internal tube surface. As the specimen was weighed to the nearest 0.1 mg, the measurement of the internal tube diameter constituted the largest source of error in the determination.

An examination of the weight of scale removed from the six specimens of a single test section showed that the scale deposit was heavier in the center of the tube than near the ends. This is attributed to heat leakage from the ends of the test tube in spite of the electric heaters which extended well beyond the ends of the section and the uniform external surface temperatures as indicated by the thermocouples. By careful manipulation of the heaters, it was possible to hold the maximum variation in scale formation in any one specimen to within 6 per cent of the mean for the entire tube.

Inasmuch as duplicate check tests showed satisfactory agreement on the basis of scale removed from successive tubes subjected to presumably identical operating conditions, a series of tests was run to determine the effect of temperature and pressure on the oxidation of low-carbon steel tubes in contact with steam. After the steam and tube were at the desired operating conditions, the steam was admitted to the tube and 30 min was allowed for adjustment and stabilization of conditions. Then readings were taken over a period of 36 hr after which the tube was removed quickly and filled with nitrogen and another tube substituted for it. With steam temperatures as low as 800 F, the 36-hr test gave a layer of scale thick enough to permit accurate weighing and was, therefore, adopted as a standard length of test. While the rate of reaction decreases with time as the layer of corrosion products increases in thickness, experience indicates that the layer of black iron oxide Fe_3O_4 which is formed under these conditions will not protect the parent metal from further oxidation. It is believed that these short-time tests give a reliable picture of the extent to which a clean metal surface is subject to oxidation by steam under various conditions of pressure and temperature.

Results of Tests on Low-Carbon Steel

In Table 2 are tabulated the results of a series of tests at a

TABLE 2 EFFECT OF TEMPERATURE UPON THE OXIDATION OF LOW-CARBON STEEL BY STEAM

Test no.	Temperature, F — Steam	Temperature, F — Metal	Steam pressure, lb per sq in., gage	Scale formed in 36 hr, g per sq in. of internal tube surface
9	801	794	1195	0.0101
7	904	894	1200	0.0125
16	1005	1001	1200	0.0370
15	1026	1023	1200	0.0452
11	1055	1045	1199	0.0432
14	1073	1072	1200	0.0642
18	1099	1097	1200	0.0973
19	1105	1099	1200	0.1065
17	1200	1195	1201	0.2970[a]

[a] 23-hr test.

constant steam pressure of 1200 lb gage with steam and metal surface temperatures from 800 to 1200 F. The high-temperature test was terminated at the end of 23 hr because of the failure of a stop valve which had been subjected to rapid fluctuations in temperature when cutting in and shutting off steam from the test sections. These data are plotted in Fig. 7 and may be represented between 800 and 1100 F by the equation

$$W = 5.56 \times 10^{-7} e^{0.011t} \qquad [2]$$

where

W = scale formed per square inch of internal tube surface in 36 hr, g

t = surface temperature, F

While the position of the curve above 1100 F is questionable because of the short test at 1200 F, it is probably steeper than indicated in Fig. 7 and the great influence of temperature upon the corrosion of low-carbon steel by steam is clearly indicated.

The slope of this curve at the high-temperature end emphasizes the importance of maintaining, in high-temperature superheater design and operation, proper steam distribution and reasonable heat-transfer rates in order that the temperature of the internal surface of the superheater tube may not become excessive.

At a constant temperature of 1100 F, steam pressure apparently has no influence on the corrosion of low-carbon steel as shown by the data in Table 3. A test at 800 lb pressure was discarded because valve trouble interferred with regulation and resulted in low temperatures. A test at 1600 lb pressure was terminated by failure of a gasket in the boiler water level-gage glass. Lack of time prevented a repetition of these tests.

To determine the possible effect of velocity at a constant metal surface temperature, a core of $^3/_4$ in. round 18-8 stainless steel 17 in. long was centered by pins in the second half of a

TABLE 3 EFFECT OF STEAM PRESSURE UPON THE OXIDATION OF LOW-CARBON STEEL BY STEAM

Test no.	Steam pressure, lb per sq in., gage	—Temperature, F— Steam	Metal	Scale formed in 36 hr, g per sq in. of internal tube surface
10	400	1100	1097	0.0962
21	397	1098	1095	0.1046
18	1200	1099	1097	0.0973
19	1200	1105	1099	0.1065

3-ft test section of low-carbon steel having an internal diameter of 1 in. The entrance end of the core was parabolic in cross section. The test section was operated at 1200 lb pressure and 1100 F for a period of 36 hr after which the scale on that part of the tube containing the core was compared with the scale on the section which did not have a core. The results are as follows:

Without core: 0.111 g of scale per sq in. of internal tube surface.

With core: 0.171 g of scale per sq in. of internal tube surface.

The scale formed in that part of the tube which did not have a core was about 8 per cent above that indicated by Fig. 7.

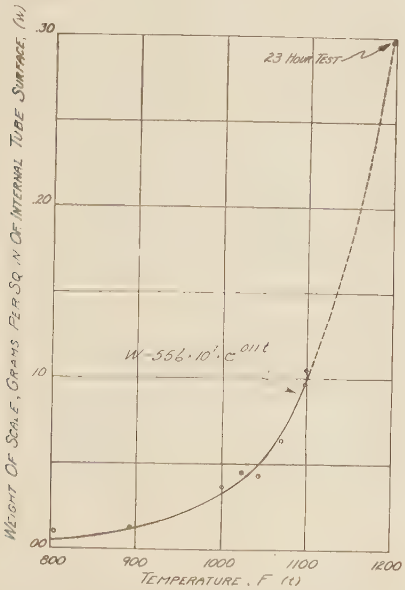

Fig. 7 Effect of Temperature on the Weight of Scale Formed
(The curve is based on 36-hr tests of low-carbon-steel tubes which were subjected to a steam pressure of 1200 lb per sq in.)

This may be due to experimental errors or possibly to flow disturbances caused by the core in the second half of the tube. The data would indicate that, at least within certain limits, increased velocity adjacent to the surface of the metal accelerates the rate of corrosion at a constant metal temperature. This matter is being investigated in more detail.

Low-Pressure Equipment

Inasmuch as experience had indicated that at 1200 lb pressure and 1000 F, steam reacts with 18-8 stainless steel stabilized with columbium and with 25-20 stainless steel to form a black-oxide coating on the tube, and it was further known that 18-8 stainless steel had stood up for at least 36,000 hr of high-temperature service, mostly at 1100 F, in the experimental superheater of The Detroit Edison Company, it was believed that the protective characteristics of the oxide layer and its ability to remain impervious under temperature fluctuations are of fundamental importance. It was decided, therefore, to construct an apparatus in which seven different tubes could be subjected simultaneously to identical steam conditions which could be held constant automatically for a considerable period of time or changed rapidly at will. In view of the apparent absence of pressure effects and the expense of operating a high-pressure boiler, it was decided to superheat steam from the regular laboratory steam header where the normal pressure is 200 lb gage.

Steam for this study was superheated to about 620 F in a gas-fired superheater from which it flowed through a coil consisting of 70 ft of $^3/_8$-in. standard iron-pipe-size seamless-steel tubing and 45 ft of $^5/_8$-in. OD $\times$ $^3/_8$-in. ID 25-20 stainless-steel tubing. This coil is capable of superheating at least 40 lb of steam per hour to 1200 F, the heating being accomplished electrically by passing alternating current through the tube itself, which is connected across the secondary of a transformer. A Leeds & Northrup recording potentiometer is connected to a thermocouple at the superheater outlet and is used to operate a small relay which in turn operates a circuit breaker in the primary circuit of the transformer. This control is capable of holding the mean steam temperature constant with cyclical variations of not over $\pm$ 7 F. Six tubes, each 5 ft long, of $1^1/_2$-in. OD $\times$ 1-in. ID material, are nested in a bundle around a seventh tube of the same size. Each tube is provided at each end with a gasket and screwed reducer of 18-8 stabilized with columbium. On the inlet end, each tube is connected by a short $^1/_4$-in. pipe nipple of stabilized 18-8 stainless steel to a common distribution flange of similar material which is bolted to a flange on the hot end of the superheater coil. The distribution flange also contains openings for a pressure gage, thermocouple, and steam sampling line. On the discharge end, each tube is connected by an 18-8 stainless-steel nipple to a seamless tubular copper condenser which is totally submerged in a tank of cold running water. The condensate flows upward through a dirt trap and $^1/_4$-in. needle valve to a gas separator. The gas separator, similar in principle to the one shown in Fig. 2, is arranged to boil the condensate and remove the gases from eight condensate lines. The gases are collected and analyzed in the same manner as in the high-pressure setup. The tube bundle, consisting of a nest of seven tubes, is covered with mica, chromel resistance wire, thermocouples, and insulation, all arranged so that the tubes are heated uniformly to any desired temperature up to at least 1200 F. In operation, the tubes and steam are maintained at the same temperature.

Results From Low-Pressure Tests

Two low-carbon steel tubes and five other tubes of the compositions indicated in Table 4 were arranged in a horizontal bundle with tube No. 7 in the center and the others surrounding it. These tubes were maintained at a uniform temperature of 1100 F for two weeks, during which they were supplied with steam also at a steady temperature of 1100 F. The gas in the steam leaving each tube was collected and analyzed and the gas content of the steam at the entrance to the tubes was also determined. The flow rate through each tube was 4.8 lb per hr.

At the end of 14 days of continuous operation, the equipment was shut down, 18 in. of tubing was cut from the discharge end of each of the seven specimens and the tubes, now 42 in. long, were reassembled and subjected to 7 days of continuous operation at 1100 F base temperature, which included 34 hr of steady temperature operation at 1100 F, followed by 10 successive swings in temperature during each of which the temperature was suddenly increased to 1150 or 1200 F, quickly reduced to from 650 to 450 F and returned to a steady temperature of 1100 F.

At the end of the 14-day period, the scale in each of the 18-in. samples was determined by the method described in connection with the high-pressure tests. The results are presented in Table 4 and are designated as Test No. 1. At the end of the period of extreme-temperature fluctuations, the scale in the remaining

TABLE 4 COMPARATIVE OXIDATION OF SIX DIFFERENT STEELS IN CONTACT WITH STEAM AT 1100 F AND 175 LB GAGE

		Analysis of tube material							Weight of scale formed			
									G per sq in. of internal tube surface		In per cent of scale formed in low-carbon steel	
Tube no.	Material	C	Mn	Si	Cr	Mo	P	S	Test No. 1	Test No. 2	Test No. 1	Test No. 2
3	Cold-drawn steel	0.10–0.20	0.30–0.60	..	..	..	0.045	0.055	0.218	0.524	100	100
7	Cold-drawn steel	0.10–0.20	0.30–0.60	..	..	..	0.045	0.055	0.205	0.520	..	..
1	Silicon-molybdenum	0.09	0.15	1.43	..	0.50	0.018	0.019	0.140	0.343	66	66
2	Timken (DM)	0.11	0.43	0.62	1.25	0.55	0.01	0.013	0.143	0.340	68	65
4	Carbon-molybdenum	0.13	0.48	0.128	..	0.51	0.013	0.012	0.162	0.469	77	90
5	B & W Croloy 3	0.15	0.48	0.45	3.12	0.8	0.011	0.016	.....	0.368	..	70
6	B & W Croloy 2	0.104	0.41	0.43	1.99	0.56	0.019	0.011	0.141	0.321	67	62

42 in. of tube was determined for each specimen in a similar manner and the result was reported in Table 4 as Test No. 2.

The data in Table 4 indicate that, with the exception of tube No. 4, the extent of oxidation of the various alloy steels is about two thirds that of low-carbon steel. The data would indicate a higher oxidation rate during the period of fluctuating-temperature operation than during the period of steady-temperature operation. However, too much reliance must not be placed on this conclusion because the data were obtained from the first test run made with the low-pressure equipment and it was found that the scale deposit was heavier in the center of each tube than near the ends, due perhaps to unequal temperatures along the tube bundle or to the influence on velocity of sudden changes in cross section at each end. Nevertheless, the seven tubes were operated under practically identical conditions as between tubes so that the results from Test No. 1 are comparable one with another, and the same is true of the results from Test No. 2.

Difficulty was encountered in machining some of the 4-in. long specimens into which the tubes were cut prior to weighing them because of the chipping of the relatively thick layer of brittle scale. The data from tube No. 5 on Test No. 1 were discarded for this reason. Also, the action of the inhibitor and solvent on the alloy steels and the relatively thick layer of scale introduced some uncertainty. Accordingly, attention is being given to the possibility of chemical removal of the oxidation products from an entire tube without cutting or machining the external surface, and a measurement of the corrosion products by titration methods.

As in the high-pressure tests, the gas-evolution data could not be correlated with the data in Table 4 and the results from the two low-carbon-steel tubes differed widely on the basis of total gas evolution, constituents, and rates. During the second or fluctuating-temperature period, the rate of gas evolution showed no correlation with the temperature variations. In the opinion of the authors, the gas-evolution data collected during a test are of little value except as a rough indication of the extent to which the layer of corrosion products is retarding the rate of attack, and, therefore, have not been reproduced in this report.

After certain changes have been made in the equipment and procedure as indicated by the experience obtained to date, the low-pressure apparatus will be used to compare the extent of oxidation of the various steels listed in this paper.

The investigation has been made possible by the cooperation with Purdue University, of the A.S.M.E., the Engineering Foundation, The Superheater Company, The Babcock & Wilcox Company, Utilities Coordinated Research, The Detroit Edison Company, National Tube Company, Crane Company, and The Timken Roller Bearing Company.

The authors also wish to acknowledge their special appreciation of the assistance rendered by Dr. T. DeVries and Dr. C. J. Klemme of the staff of Purdue University, W. H. Armacost of the Superheater Company, H. J. Kerr of The Babcock & Wilcox Company, and C. H. Fellows of The Detroit Edison Company.

Air Conditioning of Railroad Passenger Cars

By L. W. WALLACE[1] and G. G. EARLY, JR.,[2] CHICAGO, ILL.

The authors sketch briefly the development of railroad air conditioning and outline the methods of refrigeration that are used for cooling passenger cars. They give the results of a study of passenger-car air conditioning conducted in 1936 by the Association of American Railroads, which included: (*a*) Determination of the efficiency of the air-conditioning systems in use, by means of laboratory tests; (*b*) determination of the mechanical efficiency of the drive mechanisms; (*c*) a survey of the performance of the equipment in service; and (*d*) determination of the comparative costs of air conditioning. Based upon these tests, the authors develop the horsepower demand upon a locomotive by the different air-conditioning systems. They give formulas by which the cost of air conditioning per 1000 car-miles for the various systems may be calculated. These formulas take into consideration the variables confronted in actual operating service, namely, the length of the cooling season, the speed of the train, the total miles traveled per year, the cost of coal, water, ice, etc., and the proportion of time the equipment is operating and not operating during the cooling season. Tables and curves are given showing the total cost of the different methods of air conditioning for a variety of operating conditions.

History of Railroad Air Conditioning

The ventilation of railroad passenger cars for the purpose of providing patrons with comfortable and healthful atmospheric conditions has always been given much attention by railroad managements and equipment designers. Attempts were made eighty-five years ago to secure better ventilation in passenger cars by installing door-like windows which opened outward toward the back of the car. With these windows open and the train in motion, the aspirating effect of the outside air caused air to be drawn out of the car. The air leaving the car through the windows was constantly replaced by outside air entering through orifices in the car roof.

The Atchison, Topeka, and Santa Fé Railway was the pioneer among American railroads in attempting air cooling. In 1914 it equipped fifteen dining cars with air washers which washed the air with sprays of ice-cooled water. The air washers cooled the cars a few degrees, but did not have sufficient capacity to overcome the extreme desert temperatures in which the cars operated.

In the summer of 1929 the Baltimore and Ohio Railroad equipped an experimental air-conditioned coach. This was the direct forerunner of the first completely air-conditioned car, a Baltimore and Ohio diner, the *Martha Washington*, placed in service in May, 1930, between Washington, D. C., and New York City. In July of the same year, the Atchison, Topeka, and Santa Fé Railway also had in service its first air-conditioned dining car. Since that time, the number of installations has mounted until at the present time there are approximately 10,000 air-conditioned cars operated by railroads in the United States and Canada.

Methods of Refrigeration Used on Railroad Passenger Cars

There are three general types of refrigerating systems used on air-conditioned passenger cars: (1) the ice-activated system, (2) the steam-ejector system, and (3) the mechanical compression system. The mechanical compression system is divided into three classes according to the source of power for operating the compressor. These three classes are:

(*a*) The electromechanical compression system, which obtains power from a generator driven by the car axle and from a large capacity (1000 amp-hr) storage battery to operate the compressor motor.

(*b*) The direct mechanical compression system, which obtains power directly from the car axle by means of a mechanical drive. The speed of the compressor is regulated by an electric speed control which permits slippage at high car speeds.

(*c*) The internal-combustion engine, mechanical compression system, which obtains power from a propane-driven internal combustion engine and which is, consequently, independent of the car axle as a source of power. The propane is stored in tanks under the car.

Ice-Activated System. The ice-activated method of refrigeration as applied to air-conditioned railroad passenger cars uses the cold water from melting ice for cooling the air within the car. In some equipment the air is further cooled and washed by spraying cold water into the air stream. After the water has circulated through the system it is returned to an ice bunker where its temperature is reduced by the ice.

The melting ice creates an excess of water in the system which must be removed. This is accomplished by means of an overflow trap which maintains a constant water level in the ice bunker. To obtain the greatest efficiency from an ice-activated system the water that overflows should be at as high a temperature as possible. For this reason, the water discharged is always that returning from the cooling coils, which is at a slightly higher temperature than that in the bunker. In some cases, before being discharged, the waste water is circulated through an additional set of coils by means of which the incoming fresh air is precooled before passing through the regular cooling coils. This method is the most economical.

Steam-Ejector System. The steam-ejector method of re-

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[2] Research Engineer, Division of Engineering Research, Association of American Railroads. Mr. Early was graduated from Washington University, with a degree in chemical engineering in 1932. He held positions as research chemist for the Owens-Illinois Glass Co. and the Buffalo Cold Storage Co. before assuming his present duties in January, 1936.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

frigeration as applied to air-conditioned railroad passenger cars is based upon the fact that water under reduced pressures will boil at low temperatures. If the pressure is reduced to 29.6 in. of vacuum, water will boil at 50 F. When water boils it extracts heat from its surroundings equivalent to its latent heat of vaporization. Since it is possible with a steam-ejector to obtain a very low pressure, or a high degree of vacuum, and a correspondingly low temperature, this method of refrigeration is applicable to a passenger car when sufficient steam to operate the ejector is available from a locomotive boiler. The operation of the steam-ejector system is:

1 Water, at a low temperature and at approximately atmospheric pressure, is circulated through cooling coils where it absorbs heat from the mixture of recirculated air from the car and outside air.

2 After passing through the cooling coils, the water is sprayed into an evaporator where the pressure is maintained very much below that of the atmosphere. Due to the lowered pressure, the heat absorbed by the water causes a small part of it to evaporate and the temperature of the water is reduced.

3 To maintain the low pressure in the evaporator, the water vapor is removed as fast as it is developed. This is accomplished by the steam ejector.

4 The mixture of water vapor and steam is discharged into a condenser where it is condensed.

5 A purge ejector, operated by pumping through it the condenser spray water, removes the condensate and maintains the necessary low pressure in the condenser.

6 Water must be supplied to the evaporator in an amount equivalent to the vapor removed. Part of the condensate from the condenser is used for this purpose.

Mechanical Compression System. When a volatile liquid is permitted to vaporize, a reduction in temperature takes place. If the volatile liquid or refrigerant is in a closed cycle and the resultant vapor is again liquefied by the application of external work, continuous cooling may be obtained by the repeated use of the same refrigerant. The refrigerant used in railroad air-conditioning systems is dichlorodifluoromethane, commercially known as Freon, or F_{12}. The operation of the mechanical compression system is:

1 The refrigerant in a liquid state and at high pressure in a receiver is forced by its pressure through an expansion valve.

2 The expansion valve controls the flow of the liquid refrigerant from the receiver to an evaporator where a low-pressure exists. Due to the reduction in pressure, a portion of the refrigerant evaporates, thereby lowering its temperature.

3 As the low-temperature refrigerant flows through the evaporator it absorbs heat from the mixture of air from the car and outside air passing over the coils of the evaporator. This absorption of heat vaporizes the refrigerant.

4 The refrigerant vapor flows to the suction side of a compressor where it is compressed. Due to the application of external work, both the temperature and pressure of the refrigerant is raised.

5 The refrigerant vapor is discharged from the compressor into a condenser where heat is transferred from the refrigerant to the cooling medium passing over the condenser coils, thus condensing the refrigerant to a liquid state. The cooling medium of the condenser is either outside air or a combination of air and water.

6 The liquid refrigerant flows from the condenser into the receiver, after which the cycle is repeated.

Railroad Air-Conditioning Research

In order that the railroads might select the best equipment for their respective requirements, the need for an economic and engineering appraisal of the different systems became apparent. During the period of rapid development in design and growth in the number of applications of air conditioning, problems also arose pertaining to the performance of the various appurtenances which go to make up a completely air-conditioned car. Accordingly, early in 1936, a comprehensive research program was launched by the Division of Equipment Research of the Association of American Railroads. The objectives of the program were: (*a*) Determination of the efficiency of air-conditioning systems by means of laboratory tests. (*b*) Determination of the mechanical efficiency of the drive mechanisms. (*c*) Survey of the performance of the equipment in service, both by road tests and by laboratory hot-room tests. (*d*) Determination of the comparative costs of air conditioning on the basis of 1000 car-miles.

Laboratory Tests of Air-Conditioning Systems. Each of the major air-conditioning systems in use on passenger cars was put through a series of rigorous floor tests under controlled conditions representing the normal range of temperatures and resistance to the flow of air through the cooling coils that would be encountered in actual operation. The test procedure and instruments used conformed to the "Standard Method of Rating and Testing Air-Conditioning Equipment," prepared by the Joint Committee on Rating Commercial Refrigerating Equipment and sponsored by the American Society of Refrigerating Engineers.

Each air-conditioning system was subjected to nine tests with temperatures and external air-resistance pressures as shown in Table 1.

TABLE 1 CONDITIONS OF TESTS OF AIR-CONDITIONING SYSTEMS

Test number	External air-resistance pressure, in. of water	Temperature of air in Deg F: Entering condensers		Entering cooling coils	
		Dry bulb	Wet bulb	Dry bulb	Wet bulb
101	0.00	80	68	80	67
102	0.00	90	72	80	67
103	0.00	100	76	80	67
104	0.00	110	80	80	67
105	0.50	90	72	80	67
106	0.85	90	72	80	67
107	0.85	100	76	80	67
108	0.85	110	80	80	67
109	0.00	95	75	95	75

1 Four tests were run at zero external air-resistance pressure with condenser-air temperatures of 80, 90, 100, and 110 F.

2 One intermediate test was run at 0.5 in. of water external air-resistance pressure and 90 F condenser-air temperature.

3 Three tests were run at 0.85 in. of water external air-resistance pressure with condenser-air temperatures of 90, 100, and 110 F.

4 During all these tests the temperature of the air entering the cooling coils was maintained at 80 F dry bulb and 67 F wet bulb.

5 During one test of each unit, as shown by test 109, the temperature of the air passing over both the condensers and cooling coils was 95 F dry bulb and 75 F wet bulb. This test was made to determine the performance of the system under a high load, a condition similar to that experienced in precooling a hot passenger car.

The refrigerating capacity of the air-conditioning systems was measured by the quantity of air circulated and the degree of cooling. The power consumption of the air-conditioning systems was measured as the electric input to the motors operating the equipment.

Table 2 shows the refrigeration developed and the power consumption of the air-conditioning systems tested at one of the test conditions, namely, at 90 F outside-air temperature and 0.0 in. of water external air-resistance pressure. These conditions are those commonly used by the manufacturers in rating their equipment for commercial purposes.

TABLE 2 POWER CONSUMPTION OF DIFFERENT RAILROAD AIR-CONDITIONING SYSTEMS

System	Manufacturer's rated capacity, tons	Refrigeration delivered at test conditions, tons	Power consumption: Total, kw	Power consumption: Kw per ton of refrigeration
A	2.75	2.77	7.4	2.67
B	3.5	3.40	5.91	1.74
C	5	6.22	8.39	1.35
D	6	5.61	10.6	1.89
E	6	6.31	8.96	1.42
F	6	6.90	9.58	1.39
G	7	7.08	12.88	1.82
H	7	7.48	11.29	1.51
I	7	8.00	11.35	1.42
J	7	7.33	0.81[a]	0.11[a]
K	7	7.49	1.27[b]	0.17[b]
L	6	5.9	3.42[c]	0.58[c]
M	7	7	1.19[d]	0.17[d]

[a] Additional consumption of 7.55 lb of propane per hr, or 1.3 lb per hr per ton of refrigeration.
[b] Additional consumption of 7.25 lb of propane per hr, or 0.97 lb per hr per ton of refrigeration.
[c] Additional consumption of 180 lb of steam per hr, or 30.5 lb per hr per ton of refrigeration.
[d] Additional consumption of 536 lb of ice per hr, or 76.7 lb per hr per ton of refrigeration.

Laboratory Tests of Drive Mechanisms. To transfer sufficient power from the car axle to operate the air-conditioning systems some form of drive mechanism is required. The mechanical efficiencies of six different drives were obtained by measuring the power required to operate the drive at different loads. The loads were applied by a prony brake connected to the drive shaft. The power input was measured by means of a torque arm attached to the housing of an electric dynamometer which was used as a variable-speed motor. The knife-edge on the end of the torque arm rested on a platform balance where the input torque was measured. The loads applied were equivalent to car-generator outputs ranging from zero to 20 kw, at equivalent train speeds ranging from 15 to 90 mph. Table 3 shows the mechanical efficiency of six different car-axle drives for railroad air-conditioning systems at different car speeds.

TABLE 3 MECHANICAL EFFICIENCIES OF CAR-AXLE DRIVES FOR RAILROAD AIR-CONDITIONING SYSTEMS

Drive	Efficiency in per cent at car speeds of: 30 mph	50 mph	70 mph	90 mph
A	77.0	69.5	47.0	34.0
B	98.3	95.0	92.5	90.9
C	93.5	92.5	91.0	89.5
D	92.5	91.3	89.5	88.5
E	90.2	93.0	89.5	87.0
F	92.5	87.5	84.5	81.5

Drive *A* is a direct mechanical type. Each of the other drives is an electromechanical type, requiring the use of a generator and motor to transmit the power from the mechanical element of the drive to the flywheel of the compressor. The efficiency of power transmission from the car axle to the compressor flywheel is shown in Table 4.

TABLE 4 EFFICIENCIES OF POWER TRANSMISSION FROM CAR AXLE TO COMPRESSOR FLYWHEEL

Drive	Efficiency in per cent at car speeds of: 30 mph	50 mph	70 mph	90 mph
A	77.0	69.5	47.0	34.0
B	61.5	60.5	59.0	56.0
C	58.5	59.0	57.5	55.0
D	58.0	57.5	56.0	53.5
E	56.5	59.0	56.0	53.0
F	57.0	54.5	52.0	49.0

Survey of Performance of Equipment in Service

When the air-conditioning study was inaugurated, there were approximately 6000 air-conditioned cars in service. In order to round out a complete program, it was necessary to determine the performance of air-conditioned cars both under controlled conditions and in actual operating service.

For this reason tests were made of (1) 14 air-conditioned cars in a laboratory hot room, and (2) 594 cars in actual service throughout the country.

(1) *Laboratory Hot-Room Tests of Air-Conditioned Cars.* The most important results arising out of the laboratory hot-room tests of air-conditioned cars consisted of information concerning (*a*) the insulating value of the cars, (*b*) the effect of sun upon cooling requirements, and (*c*) the influence of duct design and location on air movement and temperature distribution.

(*a*) To determine the effectiveness of the insulation, heat was added by electric heaters to the inside of the car at a uniform and measured rate. After constant temperature conditions had been established the heat loss through the car structure was necessarily equal to the heat added. Under still air conditions, the inside and outside temperatures were measured to determine the temperature difference causing the heat loss.

The average coefficient of heat transmission for the roof, walls, floor, and ends, exclusive of the glass, for the 14 cars tested in the hot room, ranged from 0.263 to 0.555 Btu per hr per sq ft per deg F difference in temperature. A significant fact disclosed by these measurements is that the measured coefficient is approximately one and one-half times the coefficient calculated from a typical car section.

The heat transmission through structural-steel beams is usually not included in calculated coefficients. The tests of air-conditioned cars show that the heat loss through these beams, if not taken into account, is sufficient to cause a considerable error in the over-all coefficient of heat transmission.

(*b*) The heat load due to solar radiation was determined by measuring the refrigeration delivered by the air-conditioning equipment with sun effect present and subtracting from it the heat load without sun effect. To provide for the sun load for a 70-ft car, about 1.2 tons of additional refrigeration are required.

(*c*) Air movement and temperature distribution were measured in the car when the equipment was in continuous operation, and maintaining 80 F dry bulb and 50 per cent relative humidity at the recirculated-air grille. Air movement was measured with a Kata thermometer at each end and the center of the car at heights of 11 in. and 46 in. from the floor. Temperature measurements were made at eight locations in the car, equally spaced, and at three heights from the floor at each location, namely 11 in., 46 in., and 65 in. These three heights were used as representative of ankle level, head level when seated, and head level when standing, respectively.

The effectiveness of each air-delivery system was measured in terms of (*a*) the uniformity of temperature throughout the car, (*b*) the velocity and uniformity of the air movement, (*c*) the appearance or nonappearance of hot or cold spots, (*d*) the presence or absence of drafts, and (*e*) the characteristics of the air flow. In terms of the foregoing factors, the several types of air delivery fall in the following order of preference:

1 Inside center duct, with approximately 20 outlets.

2 Double outside ducts, with the outlets in each duct not directly opposite those of the other duct.

3 Single outside duct. This type is much less desirable than either of the two foregoing.

4 Double end bulkhead.

5 Center bulkhead.

Tests were recently made on one car using a multivent method of air delivery which has approximately 2000 small circular openings per sq ft of ceiling area. The temperature distribution obtained on this car was superior to that obtained on any of the 14 cars tested. This type of air distribution, however, is essentially the same as the inside center duct, with thousands of outlets.

(2) *Road Tests of Air-Conditioned Cars in Service.* Through-

out the entire country and on thirty-one different railroads tests were made on 594 cars in actual operation. The road tests disclosed two important facts: First, that more adequate and positive control of temperature and humidity should be provided; and second, that many of the air-conditioning systems are not supplying sufficient outside air. An insufficient supply of outside air is one of the frequent causes of odors in air-conditioned enclosures.

Determination of Cost of Air Conditioning. The finance, accounting, taxation, and valuation department of the Association of American Railroads obtained from the accounting departments of the sixteen railroads which were foremost in passenger-car air conditioning the necessary information for the determination of the investment and maintenance costs of air conditioning. When included with the operation costs, which were determined by the results of the equipment tests, the total cost of air conditioning was derived.

Power Demand Upon a Locomotive by Different Air-Conditioning Systems

Power Required to Operate the Air-Conditioning Equipment. The average amounts of electrical and mechanical energy, steam, ice, and propane consumed by the various systems when in continuous operation at average temperatures are known for the laboratory-test data.

(a) Electromechanical........	10.5 kw
(b) Direct-drive, mechanical...	1.0 kw and 10.24 hp
(c) Internal-combustion engine, mechanical.............	1.23 kw and 7.3 lb propane per hr
(d) Steam-ejector............	7.2 hp and 230 lb steam per hr
(e) Ice-activated.............	1.61 hp and 463 lb ice per hr

Horsepower Required at Car Axle to Operate Air-Conditioning Equipment. The horsepower required at the car axle to furnish the above mechanical and electrical power is higher due to the losses in the driving mechanism at the speed of train operation. For the purpose of the calculations which follow, speeds of 30, 50, 70, and 90 mph have been chosen. The efficiency of the driving mechanisms at these speeds is shown by the following:

	Mechanical efficiency in per cent at car speeds of			
	30 mph	50 mph	70 mph	90 mph
Efficiency of direct drive	77.5	70	47	34.5
Average efficiency of four mechanical drives and generators (for electric power generation)	75	75.4	73	70

The horsepower required at the car axle is accordingly: $hp_a = hp_b/E_1$

Where

hp_a = horsepower required at car axle

hp_b = horsepower required to operate air-conditioning equipment

E_1 = efficiency of drive in per cent

Horsepower Required at Drawbar to Operate Air-Conditioning Equipment. The drawbar horsepower is higher than that at the axle due to axle-bearing friction. A friction loss of 5 per cent has been assumed. Therefore, the horsepower at the drawbar is $hp_{db} = hp_a/0.95$.

Horsepower Required at Drawbar to Haul Added Weight of Air-Conditioning Equipment. In addition to the drawbar horsepower necessary for the mechanical operation of the equipment, power is required to haul the added weight of the equipment. The average increase in weight of cars due to air conditioning is:

Electromechanical..........................	4.8 tons
Direct mechanical..........................	4.3 tons
Internal-combustion engine, mechanical........	4.3 tons
Steam-ejector................................	5.65 tons
Ice-activated................................	4.25 tons

From published data[3] on passenger-train resistance, the calculated drawbar horsepower required for each additional ton of weight is as follows:

	Car speeds, mph			
	30	50	70	90
Drawbar horsepower required per ton of added weight	0.24	0.53	1.06	2.16

Multiplication of the weights of the air-conditioning equipment in tons by the above factors will give the additional drawbar horsepower required to haul the weight of the equipment at the various speeds.

Total Power Demand Upon a Locomotive by One Air-Conditioned Car. The total power demand upon the locomotive by the different types of air-conditioning systems at a given speed is accordingly the sum of the power required to operate the equipment and the power required to haul the weight of the equipment at that speed. Table 5 shows the total drawbar horsepower per car required by the different air-conditioning systems for speeds ranging from 30 to 90 mph.

TABLE 5 TOTAL DRAWBAR HORSEPOWER REQUIRED PER CAR BY DIFFERENT AIR-CONDITIONING SYSTEMS WHEN IN CONTINUOUS OPERATION

	Drawbar hp at car speeds of			
Type of air-conditioning system	30 mph	50 mph	70 mph	90 mph
Electromechanical..............	20.9	22.2	25.4	31.6
Direct mechanical..............	16.8	19.5	29.5	42.6
Internal-combustion engine, mechanical..............	3.4	4.6	7.0	11.9
Steam-ejector[a]..............	7.6	9.3	12.5	19.0
Ice-activated..............	3.2	4.5	6.8	11.6

[a] In addition, power from the locomotive is required to the extent of 230 lb of steam per hr.

TABLE 6 TOTAL DRAWBAR HORSEPOWER REQUIRED PER CAR BY DIFFERENT AIR-CONDITIONING SYSTEMS WHEN NOT IN OPERATION

	Drawbar hp at car speeds of			
Type of air-conditioning system	30 mph	50 mph	70 mph	90 mph
Electromechanical..............	5.0	6.4	9.0	14.4
Direct mechanical..............	5.2	6.4	8.7	13.5
Internal-combustion engine, mechanical..............	2.2	3.5	5.8	10.6
Steam-ejector..............	3.7	5.4	8.4	14.8
Ice-activated..............	1.9	3.2	5.5	10.2

TABLE 7 AVERAGE DRAWBAR-HORSEPOWER DEMAND UPON THE LOCOMOTIVE PER CAR, FOR DIFFERENT AIR-CONDITIONING SYSTEMS

	Average drawbar hp at car speeds of			
Type of air-conditioning system	30 mph	50 mph	70 mph	90 mph
Electromechanical..............	13.9	15.2	18.2	24.0
Direct mechanical..............	11.7	13.7	20.3	29.6
Internal-combustion engine, mechanical..............	2.8	4.1	6.5	11.4
Steam-ejector[a]..............	5.9	7.6	10.7	17.5
Ice-activated..............	2.6	3.9	6.2	11.0

[a] In addition, power from the locomotive is required to the extent of 230 lb of steam per hr during the time the equipment is in operation.

When an air-conditioning equipment is not operating, there is still a power demand upon the locomotive due to (a) the power required to haul the added weight of the equipment, (b) the power required to operate the cooling-coil fan, (c) the power required to overcome drive and generator friction, and (d) the losses occasioned by taking power from storage batteries. Table 6 shows the drawbar horsepower required by each of the air-conditioning systems when the equipment is not operating.

From the results of many tests it was found that for average conditions the average cooling capacity of air-conditioning systems was 5.92 tons and that the average refrigeration load on a car during the cooling season was 3.3 tons. The ratio of demand

[3] Mechanical Engineers' Handbook, Lionel S. Marks, Editor-in-Chief, McGraw-Hill Book Co., Inc. (3rd ed.) 1930, p. 1489.

to capacity, $3.3/5.92 = 0.56$ or 56 per cent, is the percentage of time the cooling system is in operation, on an average, during the cooling season.

If 56 per cent of the drawbar horsepower required when an air-conditioning system is in operation is added to 44 per cent of the drawbar horsepower required when the air-conditioning system is not in operation, the sum is the average drawbar-horsepower demand upon a locomotive during the cooling season. The values of this average drawbar-horsepower demand for each of the air-conditioning systems are given in Table 7.

Cost of Railroad Air Conditioning

The total cost of railroad air conditioning consists of three essential elements: (*a*) Cost of installation, (*b*) cost of maintenance, and (*c*) cost of operation.

Cost of Installation. The cost of installing an air-conditioning system on a railroad car includes: (1) Purchase price of all the parts of an air-conditioning system; and (2) expenditure for labor, miscellaneous materials, and overhead expense required for the installation.

The gross installation cost reported by 16 railroads is shown in the following tabulation for the five types of air-conditioning systems:

	Gross installation cost
Electromechanical	\$6484.00
Direct mechanical	\$8515.00
Internal-combustion engine, mechanical	\$5750.00
Steam-ejector	\$8475.00
Ice-activated	\$3982.00

The gross installation cost may be amortized on the following bases:

(*a*)	Depreciation rate	12.5 per cent
(*b*)	Interest rate	6 per cent
(*c*)	Taxes and insurance rate	1.5 per cent
	Total	20 per cent

The fixed charges per 1000 car-miles for any type of system are: $FC = 1000 \times 0.20\, A/m$

where

FC = fixed charges in dollars per 1000 car-miles
A = gross installation cost in dollars
m = total number of car-miles traveled in one year

Cost of Maintenance. Since the air conditioning of railroad passenger cars is a relatively new undertaking, a standard method of procedure for segregating maintenance costs has not been developed. However, the cost of maintenance reported by 16 railroads is shown in the following tabulation for the five types of air-conditioning systems:

System	Number of units	Average maintenance cost per unit per 1000 car-miles
Electromechanical	391	\$3.33
Direct mechanical	312	2.33
Internal-combustion engine, mechanical	45	3.30
Steam-ejector	223	2.15
Ice-activated	527	0.97

Expressed by formula, the maintenance cost per 1000 car-miles for any type of system is $MC = 1000B/m$

where

MC = maintenance cost in dollars per 1000 car-miles
B = annual maintenance cost in dollars
m = total number of car-miles traveled in one year

Cost of Operation. The cost of operation of each air-conditioning system is influenced by the following factors:

1 Speed of Train Operation. The train speed affects the efficiency of the drive mechanism, the horsepower required to haul the weight of the equipment, and the time required to cover a given distance, which is proportional to the amount of electricity, steam, ice, or propane consumed.

2 Cost of Producing a Horsepower-Hour by the Locomotive. For average purposes, 3.5 lb of coal are consumed by a locomotive per cylinder hp and 6 lb of water are used to 1 lb of coal. With coal at an average determined price of \$2.39 per ton, water at \$0.75 per 1000 cu ft, and a locomotive mechanical efficiency of 90 per cent, the cost of coal and water for one horsepower-hour becomes:

$$\frac{3.5 \times \$2.39}{2000 \times 0.90} + \frac{3.5 \times \$0.75}{0.90 \times 1000 \times 62.4} = \$0.00493 \text{ per hp-hr}$$

3 Ratio of Operating Time to Nonoperating Time. As pointed out in the section on the horsepower demand upon the locomotive, an air-conditioning system operates on an average of 56 per cent of the time during the cooling season.

4 Length of Cooling Season. The length of the cooling season has a decided effect upon the operating cost. The actual cost of operation per 1000 car-miles for an entire year is composed of the cost of operation during the cooling season plus that cost associated with the remainder of the year.

5 Cost of Additional Necessities. At current prices, the cost of ice in a bunker may be taken as \$4.42 per ton, the cost of propane on a car as \$0.039 per lb, and the cost of steam as the actual cost of the coal and water used in generating the amount of steam required.

Considering all the variable factors, the cost of operation per 1000 car-miles for any railroad air-conditioning system may be calculated by the following formula:

$$OC = \frac{1000K}{12S}\,(D \times E + 0.56F \times G) + \frac{1000(12 - K)}{12S}\,(H \times E)$$

where

OC = operation cost in dollars per 1000 car-miles
D = average drawbar-horsepower demand on a locomotive by the air-conditioning system at speed S. (Table 7)
E = cost per hp-hr, dollars
F = additional necessities, such as ice, steam, or propane, lb per hr
G = cost of additional necessities, dollars per lb
H = drawbar hp required when system is not operating (Table 6)
S = speed of train operation, mph
K = length of cooling season, months
0.56 = proportion of operation time to total time during the cooling season

Total Cost of Air Conditioning. When fixed charges, maintenance cost, and operation cost are each expressed on the basis of 1000 car-miles, as in the foregoing, addition of the three elements will give the total cost of air conditioning upon that same basis.

From an operating standpoint, the cost of air conditioning is affected by the variable factors previously outlined. In addition, the economics of air conditioning is affected by another factor, the total number of car-miles per year, which directly influences the fixed charges per 1000 car-miles.

Because of these variables, a categorical statement cannot be made as to which system is the best from the standpoint of economics. In each case, the selection has to be made on the basis of the prevailing operating circumstances that are confronted,

TABLE 8 COMPARATIVE TOTAL COSTS PER 1000 CAR-MILES OF FIVE DIFFERENT METHODS OF AIR CONDITIONING RAILROAD PASSENGER CARS FOR A COOLING SEASON OF THREE MONTHS

Cooling season of three months	Cost per 1000 car-miles on the basis of the following car-miles per year				
	50,000	100,000	150,000	200,000	250,000
Train speed 30 mph					
Electromechanical	$29.89	$17.20	$12.96	$10.85	$ 9.58
Direct mechanical	37.47	20.46	14.80	11.95	10.25
Internal-combustion engine, mechanical	28.27	16.91	13.13	11.24	10.10
Steam-ejector	36.95	20.18	14.58	11.79	10.10
Ice-activated	21.74	14.08	11.52	10.25	9.48
Train speed 50 mph					
Electromechanical	$29.56	$16.87	$12.63	$10.52	$ 9.25
Direct mechanical	37.16	20.15	14.49	11.64	9.94
Internal-combustion engine, mechanical	27.50	16.14	12.36	10.47	9.33
Steam-ejector	36.62	19.85	14.25	11.46	9.77
Ice-activated	19.70	12.04	9.48	8.21	7.44
Train speed 70 mph					
Electromechanical	$29.50	$16.81	$12.57	$10.46	$ 9.19
Direct mechanical	37.17	20.16	14.50	11.65	9.95
Internal-combustion engine, mechanical	27.24	15.88	12.10	10.21	9.07
Steam-ejector	36.57	19.80	14.20	11.41	9.72
Ice-activated	18.88	11.22	8.66	7.39	6.62
Train speed 90 mph					
Electromechanical	$29.63	$16.94	$12.70	$10.59	$ 9.32
Direct mechanical	37.31	20.30	14.64	11.79	10.09
Internal-combustion engine, mechanical	27.23	15.87	12.09	10.20	9.06
Steam-ejector	36.73	19.96	14.36	11.57	9.88
Ice-activated	18.57	10.91	8.35	7.08	6.31

TABLE 9 COMPARATIVE TOTAL COSTS PER 1000 CAR-MILES OF FIVE DIFFERENT METHODS OF AIR CONDITIONING RAILROAD PASSENGER CARS FOR A COOLING SEASON OF FIVE MONTHS

Cooling season of five months	Cost per 1000 car-miles on the basis of the following car-miles per yr				
	50,000	100,000	150,000	200,000	250,000
Train speed of 30 mph					
Electromechanical	$30.13	$17.44	$13.20	$11.09	$ 9.82
Direct mechanical	37.65	20.64	14.98	12.13	10.43
Internal-combustion engine, mechanical	29.11	17.75	13.97	12.08	10.94
Steam-ejector	37.13	20.36	14.76	11.97	10.28
Ice-activated	24.91	17.25	14.69	13.42	12.65
Train speed of 50 mph					
Electromechanical	$29.70	$17.01	$12.77	$10.66	$ 9.39
Direct mechanical	37.28	20.27	14.61	11.76	10.06
Internal-combustion engine, mechanical	28.00	16.64	12.86	10.97	9.83
Steam-ejector	36.72	19.95	14.35	11.56	9.87
Ice-activated	21.59	13.93	11.37	10.10	9.33
Train speed of 70 mph					
Electromechanical	$29.61	$16.92	$12.68	$10.57	$ 9.30
Direct mechanical	37.30	20.29	14.63	11.78	10.08
Internal-combustion engine, mechanical	27.50	16.24	12.46	10.57	9.43
Steam-ejector	36.65	19.88	14.28	11.49	9.80
Ice-activated	20.24	12.58	10.02	8.75	7.98
Train speed of 90 mph					
Electromechanical	$29.72	$17.03	$12.79	$10.68	$ 9.41
Direct mechanical	37.46	20.45	14.79	11.94	10.24
Internal-combustion engine, mechanical	27.52	16.16	12.80	10.91	9.77
Steam-ejector	36.79	20.02	14.42	11.63	9.94
Ice-activated	19.62	11.96	9.40	8.13	7.36

which must include (*a*) the length of the cooling season, (*b*) the cooling load, (*c*) the total car-miles traveled, (*d*) the length, speed, and schedule of the runs, and (*e*) the costs of ice, fuel, and water in the locality considered.

Comparisons of the total cost per 1000 car-miles for the five methods of air conditioning are shown in Tables 8 to 11, and Figs. 1 to 4 for: (*a*) Cooling seasons of 3, 5, 8, and 10 months, (*b*) average train speeds of 30, 50, 70, and 90 mph, and (*c*) total car-mileages ranging from 50,000 to 250,000 miles per year.

A study of the cost phases of a railroad passenger-car air-conditioning system brings out several facts of prime importance. Table 12 compares the installation cost and the fixed charges, operation, and maintenance costs for the various air-conditioning systems computed for a condition fairly representative of the average for the entire country. The facts disclosed by the data in Table 12 justify the following conclusions:

1 The installation cost of air-conditioning equipment is too high. The purchase price of the equipment is 50 to 75 per cent of the installation cost, the remainder being caused by necessary changes in car construction. Reduction of the installation cost must come mainly through reduction in the purchase price of the equipment. There is nothing complicated about the railroad air-conditioning unit, merely the usual combination of heat-exchange devices, and there is no reason why these devices cannot be manufactured upon a production basis at a cost which is but a small fraction of the present cost. A mechanical compression system has no more intricate and complicated mechanisms than an automobile, yet it sells for a price of 3 to 4 times that of an average automobile.

2 The maintenance cost, except for the ice-activated system, is

TABLE 10 COMPARATIVE TOTAL COSTS PER 1000 CAR-MILES OF FIVE DIFFERENT METHODS OF AIR CONDITIONING RAILROAD PASSENGER CARS FOR A COOLING SEASON OF EIGHT MONTHS

Cooling season of eight months	Cost per 1000 car-miles on the basis of the following car-miles per yr				
	50,000	100,000	150,000	200,000	250,000
Train speed of 30 mph					
Electromechanical	$30.49	$17.80	$13.56	$11.45	$10.18
Direct mechanical	37.91	20.90	15.24	12.39	10.69
Internal-combustion engine, mechanical	30.36	19.00	15.22	13.33	12.19
Steam-ejector	37.39	20.62	15.02	12.23	10.54
Ice-activated	29.63	21.97	19.41	18.14	17.37
Train speed of 50 mph					
Electromechanical	$29.92	$17.23	$12.99	$10.88	$ 9.61
Direct mechanical	37.45	20.44	14.78	11.93	10.23
Internal-combustion engine, mechanical	28.77	17.41	13.63	11.74	10.60
Steam-ejector	36.88	20.11	14.51	11.72	10.03
Ice-activated	24.44	16.78	14.22	12.95	12.18
Train speed of 70 mph					
Electromechanical	$29.77	$17.08	$12.84	$10.73	$ 9.46
Direct mechanical	37.51	20.50	14.84	11.99	10.29
Internal-combustion engine, mechanical	28.14	16.78	13.00	11.11	9.97
Steam-ejector	36.77	20.00	14.40	11.61	9.92
Ice-activated	22.27	14.61	12.05	10.78	10.01
Train speed of 90 mph					
Electromechanical	$29.85	$17.16	$12.92	$10.81	$ 9.54
Direct mechanical	37.68	20.67	15.01	12.16	10.46
Internal-combustion engine, mechanical	27.94	16.58	12.80	10.91	9.77
Steam-ejector	36.88	20.11	14.51	11.72	10.03
Ice-activated	21.21	13.55	10.99	9.72	8.95

TABLE 11 COMPARATIVE TOTAL COSTS PER 1000 CAR-MILES OF FIVE DIFFERENT METHODS OF AIR CONDITIONING RAILROAD PASSENGER CARS FOR A COOLING SEASON OF TEN MONTHS

Cooling season of ten months	Cost per 1000 car-miles on the basis of the following car-miles per yr				
	50,000	100,000	150,000	200,000	250,000
Train speed of 30 mph					
Electromechanical	$30.73	$18.04	$13.80	$11.69	$10.42
Direct mechanical	38.09	21.08	15.42	12.57	10.87
Internal-combustion engine, mechanical	31.22	19.86	16.08	14.19	13.05
Steam-ejector	37.57	20.80	15.20	12.41	10.72
Ice-activated	32.81	25.15	22.59	21.32	20.55
Train speed of 50 mph					
Electromechanical	$30.07	$17.38	$13.14	$11.03	$ 9.76
Direct mechanical	37.58	20.57	14.91	12.06	10.36
Internal-combustion engine, mechanical	29.29	17.93	14.15	12.26	11.12
Steam-ejector	36.99	20.22	14.62	11.83	10.14
Ice-activated	27.17	19.51	16.95	15.68	14.91
Train speed of 70 mph					
Electromechanical	$29.88	$17.19	$12.95	$10.84	$ 9.57
Direct mechanical	37.64	20.63	14.97	12.12	10.42
Internal-combustion engine, mechanical	28.50	17.14	13.36	11.47	10.33
Steam-ejector	36.84	20.07	14.47	11.68	9.99
Ice-activated	23.62	15.96	13.40	12.13	11.36
Train speed of 90 mph					
Electromechanical	$29.94	$17.25	$13.01	$10.90	$ 9.63
Direct mechanical	37.83	20.82	15.16	12.31	10.61
Internal-combustion engine, mechanical	28.21	16.85	13.07	11.18	10.04
Steam-ejector	36.94	20.17	14.57	11.78	10.09
Ice-activated	22.26	14.60	12.04	10.77	10.00

TABLE 12 COST OF RAILROAD AIR CONDITIONING FOR DIFFERENT TYPES OF SYSTEMS

Type of air-conditioning system	Gross installation cost	Cost per 1000 car-miles for an average cooling season of 5 months, an average train speed of 50 mph, and an average car mileage of 150,00 miles per yr		
		Fixed charges	Operation cost	Maintenance cost
Electromechanical	$6,484.00	$ 8.65	$0.99	$3.33
Direct mechanical	8,515.00	11.35	0.93	2.33
Internal-combustion engine, mechanical	5,750.00	7.67	1.99	3.30
Steam-ejector	8,475.00	11.30	1.02	2.15
Ice-activated	3,982.00	5.31	5.29	0.97

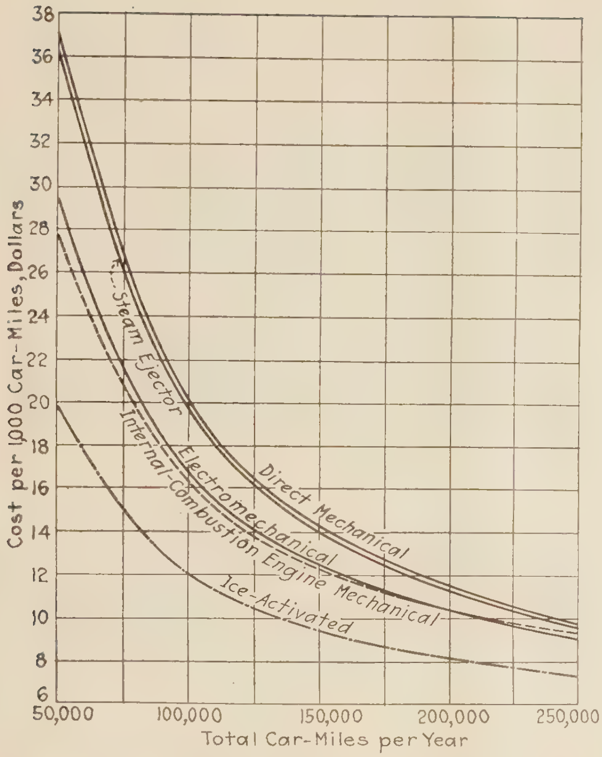

FIG. 1 COMPARATIVE TOTAL COSTS PER 1000 CAR-MILES OF FIVE DIFFERENT METHODS OF AIR CONDITIONING RAILROAD PASSENGER CARS FOR A COOLING SEASON OF 3 MONTHS, AT AN AVERAGE TRAIN SPEED OF 50 MPH

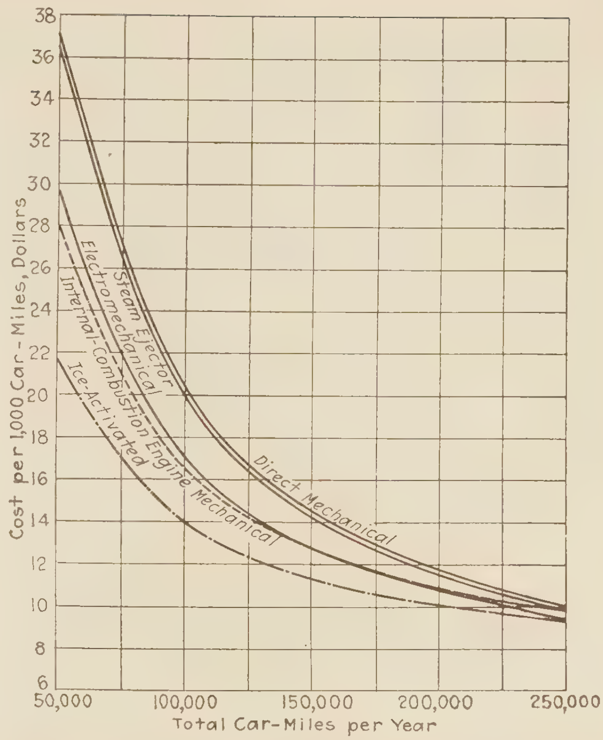

FIG. 2 COMPARATIVE TOTAL COSTS PER 1000 CAR-MILES OF FIVE DIFFERENT METHODS OF AIR CONDITIONING RAILROAD PASSENGER CARS FOR A COOLING SEASON OF 5 MONTHS, AT AN AVERAGE TRAIN SPEED OF 50 MPH

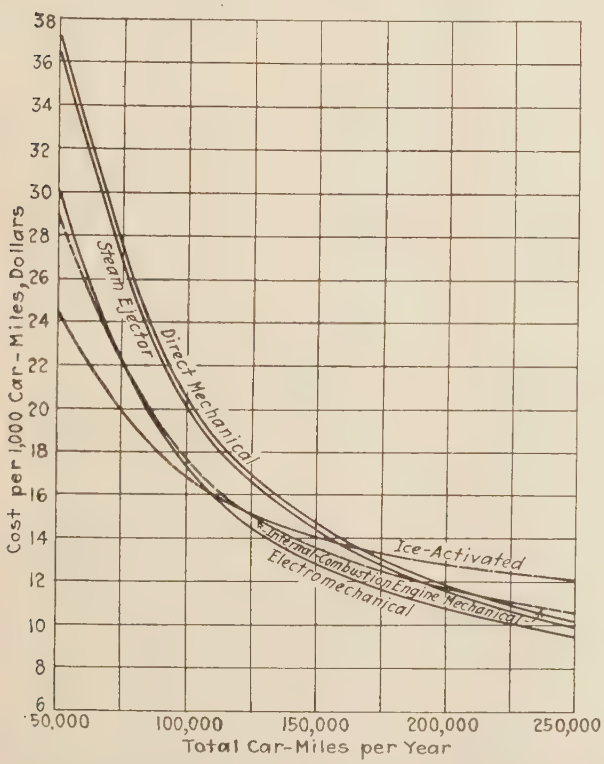

FIG. 3 COMPARATIVE TOTAL COSTS PER 1000 CAR-MILES OF FIVE DIFFERENT METHODS OF AIR CONDITIONING RAILROAD PASSENGER CARS FOR A COOLING SEASON OF 8 MONTHS, AT AN AVERAGE TRAIN SPEED OF 50 MPH

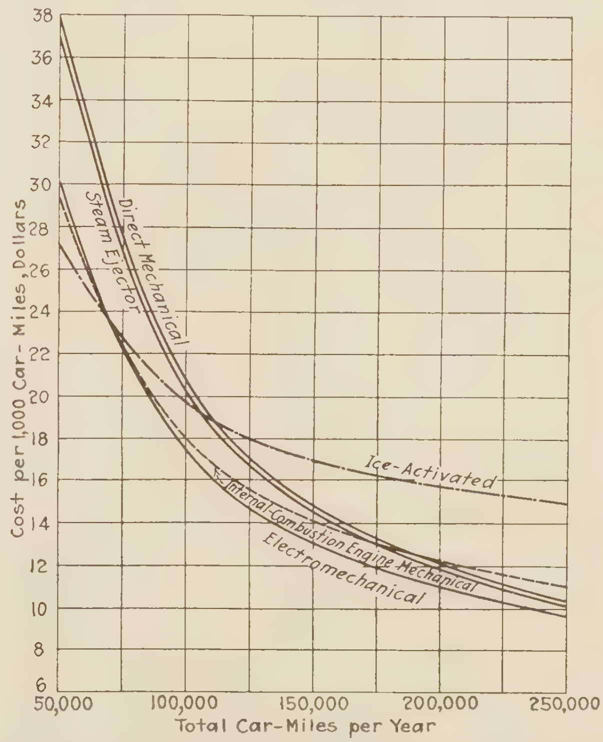

FIG. 4 COMPARATIVE TOTAL COSTS PER 1000 CAR-MILES OF FIVE DIFFERENT METHODS OF AIR CONDITIONING RAILROAD PASSENGER CARS FOR A COOLING SEASON OF 10 MONTHS, AT AN AVERAGE TRAIN SPEED OF 50 MPH

much more than the operation cost. When the premium price which is paid for the equipment is considered, the maintenance cost of an air-conditioning system should be in the same relationship with the operation cost as it is in the case of an automobile.

3 No great reduction in the operating cost is possible, except for the ice-activated system, which is dependent upon the cost of ice.

4 The fact that an air-conditioning system is the least economical from the operation standpoint does not necessarily make it the least desirable system, provided the purchase price is not excessive. The ice-activated system, with the highest operating cost, is still the most economical system to use in certain cases because of its low installation cost. It is important to note that, since the operating cost is such a small percentage of the total cost, from an economic standpoint, a difference of $100 is all that is justifiable in the purchase prices of the most efficient and the least efficient electromechanical compression systems that are suitable for railroad use. Or, conversely, a railroad air-conditioning system may be the least efficient from a power-consumption standpoint, but it can still be on the same economical basis as the most efficient system, provided its purchase price is $100 less than that of the most efficient system.

Conclusion

1 The initial investment required for the air conditioning of railroad passenger cars is too high and should be reduced. We shall be greatly interested in any plan which will mean a very substantial reduction in the initial investment.

2 Too much emphasis has been placed upon cooling, which is only one phase of air conditioning. A corresponding emphasis should be placed upon each of the five essentials: Heating, cooling, ventilation and cleaning, humidification, and dehumidification.

3 There is need for a more adequate and positive automatic control of temperature and humidity.

4 The economy and efficiency of the individual elements which constitute an air-conditioning system require attention. It is not at all clear that each of these elements performs with that degree of efficiency reasonably to be expected. Apparently there has been too much striving for wide application and not sufficient analysis of conditions to be corrected.

5 The power demand upon a locomotive by an air-conditioned car ranged from approximately 4 to 15 hp at an average speed of 50 mph. For the average train of air-conditioned cars, the power requirement for air conditioning alone may be as much as 10 per cent of the available power of the locomotive. It is clearly evident that the power demand upon the locomotive by air conditioning should be reduced.

6 There is evidence to show that it would not be in the interest of economy, efficiency, or satisfactory air conditioning to adapt a given system to all conditions. There is evidence to show that the selection of an air-conditioning system should be made on the basis of the circumstances associated with each geographical area together with investment, economy, and efficiency factors. There is no universality associated with air conditioning and no universal means of accomplishment.

Colors and Gums Used in Textile Printing

By IVAN F. CHAMBERS,[1] WILMINGTON, DEL.

The text summarizes the uses and methods of application of various types of colors employed in printing textiles. Each class of colors is discussed individually from the viewpoint of fastness, method, and mechanics of application, kind of goods it is best suited for, and its compatibility with other classes of colors.

Vat colors are particularly stressed both because of their all-round superior fastness as a class and because of their adaptability to both direct and discharge printing. An interesting feature is the description of how chemical research has extended the field of vat and insoluble azoic colors, not only in the range of colors made available, but also in the ease and simplification of their application.

THE PURPOSE of this paper is to outline briefly the more important colors and gums used in textile printing in this country. Colors and procedures which are in current use for special articles will not be discussed as the scope of a paper of this kind is necessarily limited. A large percentage of the total yardage of printed textile materials is processed, however, by well-established procedures employing a relatively few general classes of colors. Among the more important colors used for printing may be mentioned the following: (*a*) vat colors, (*b*) insoluble azo colors (stabilized azoic colors), (*c*) basic colors, (*d*) chrome colors, and (*e*) acid colors.

Vat Colors

Vat colors as a class are distinguished by their excellent fastness properties, being superior to all other classes of synthetic colors in resistance to washing, bleaching, light, and other destructive agencies. Vat colors are therefore employed for printing the highest grade of cotton, linen, and rayon fabrics including shirting material, draperies, women's and children's dress goods, and other fabrics which are subjected to maximum wear and repeated launderings.

In addition to their excellent fastness properties, vat colors are readily adaptable to the various methods or styles of machine printing. The two most important methods from the point of view of yardage printed are the direct- and discharge-printed styles. Direct printing refers to printing on white or undyed cloth while discharge printing consists in printing on a fabric which has been dyed with a dischargeable azo color.

The oldest known representative of the vat colors is indigo but this color has been superseded for printing purposes by the halogenated indigoids, which are brighter and have better fastness properties. The modern vat-color series has been augmented by derivatives of thioindigo, anthraquinone, and carbazole, so that an extensive range of shades is available today.

Vat colors in their normal state are water-insoluble pigments and are marketed in the form of a paste or a powder. The paste type is manufactured by grinding the pigment in water with a dispersing agent which prevents coagulation or agglomeration of the pigment particles. The individual particle is very small; typical pastes containing particles varying from $^1/_2$ micron to 8 microns in diameter. The powder type of vat color is prepared by mixing the dry pigment with dextrin, sugar, or other water-soluble diluent. The powders are usually mixed with water to form pastes before use.

When vat colors are treated with a reducing agent in the presence of alkali they become water-soluble and it is in this water-soluble or leuco form that impregnation of the fabric with the color takes place. A typical vat-color printing paste is prepared by incorporating the color in a vat-color printing gum which contains the ingredients necessary for reduction of the color. The printing gum is usually prepared in stock quantities and may be employed for a considerable length of time without deterioration. A normal printing gum is prepared by adding potash, reducing agent, and glycerine to an aqueous starch or dextrin paste. The reducing agent universally employed for printing vat colors is sodium sulphoxylate formaldehyde, a compound which is stable and relatively inert at room temperature but at elevated temperatures, such as are encountered in a vat-color ager or steam box, becomes strongly reducing in character.

The general procedure consists in printing the fabric with the vat-color printing paste, drying, and then aging the printed material for from 4 to 8 minutes. Aging is a term which signifies exposure of the goods to live steam, precautions being taken to prevent mixture of air with the steam. During the aging process, the dyestuff is reduced to its soluble form and thoroughly permeates the fabric. After aging, the goods are immersed for a short time in an oxidation bath of sodium bichromate or sodium perborate which oxidizes the vat color to its insoluble form, the color becoming essentially an integral part of the fiber. The goods are then rinsed and soaped to remove the residue of the printing paste and to brighten the shade of the color.

Vat colors are equally suitable for direct printing or discharge printing. Direct printing on white or undyed fabrics requires little comment. In the case of discharge printing, the fabric is dyed the desired shade with a dischargeable azo dyestuff and is then ready to be printed. Many azo colors employed for dyeing undergo degradation of the azo linkage when treated with a reducing agent and are transformed into colorless or slightly colored degradation products. This is a very convenient property as the reducing agent employed in the reduction of the vat color may be employed for the simultaneous degradation or discharge of the azo dyestuff with which the fabric has been previously dyed. Vat-color printing pastes for discharge printing necessarily contain a higher percentage of sodium sulphoxylate formaldehyde than printing pastes for direct printing as sufficient reducing agent is required to reduce the vat color and also discharge the azo-color ground shade.

Since vat colors in their normal state are water-insoluble pigments, it has naturally occurred to dyestuff chemists that soluble derivatives would be of great value, provided of course that the original vat color could be easily regenerated. The solution of this problem was a remarkable feat in dyestuff chem-

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Discussion of this paper should be addressed to the Secretary, A.S.M.E., 29 West 39th Street, New York, N. Y., and will be accepted until January 10, 1938, for publication at a later date. Discussion received after the closing date will be returned.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

istry and since 1922 a class of colors known as the indigosols or soluble vat colors has been available to the printer. The soluble vat colors are vat colors which have been converted in their reduced state to sulphuric esters, products which are readily soluble in water. In principle the regeneration of the water-soluble vat color to the normal pigment form depends on acid hydrolysis and simultaneous oxidation. Since no reducing agent is employed in the application of the soluble vat colors, these products are employed principally for direct printing, although discharge styles are theoretically possible. The soluble vat colors may be printed on both vegetable and animal fibers. At the present time a very extensive use of these colors is in conjunction with the insoluble azo colors for printing women's and children's dress wear. This procedure will be more fully discussed in the following paragraphs.

Insoluble Azo Colors

The insoluble azo colors form a very important class of colors employed for the direct printing of cotton goods. They are generally inferior to the vat colors in fastness properties but are more economical to use and therefore find a wide application particularly for women's dress goods. The range of shades is not entirely complete and the fastness properties in light shades or tints are not commercially satisfactory, but these deficiencies are obviated in a large measure by substituting the soluble vat colors in the pattern when this is necessary.

The insoluble azo colors are applied in the form of intermediates, the color pigment being formed on and within the fiber by reaction between the intermediates. The basic reaction which produces these colors is the reaction between a diazotized aromatic amine and a naphthol.

There are three general methods of applying the intermediates which produce the insoluble azo colors:

(*a*) A thickened solution of a diazo salt is printed on a fabric which has been impregnated with a naphthol.

(*b*) A mixture of a nitrosamine and a naphthol is printed on a white fabric.

(*c*) A mixture of a diazoimino compound and a naphthol is printed on a white fabric.

The first method mentioned is the original method devised for printing these colors and is the simplest from the chemical point of view. The cotton fabric to be printed is prepared or padded with an alkaline solution of a naphthol and dried. A printing paste is then prepared by dissolving a diazo salt, of which there are a number of standard brands on the market, in a neutral printing gum consisting usually of a mixture of starch and gum tragacanth. The diazo-salt printing paste is printed as soon as possible after preparation on the naphthol-prepared goods, the color being produced immediately on contact between the printing paste and the prepared fabric. After printing, the goods are dried and then soaped thoroughly to remove the excess naphthol which was not utilized to form color. If desired the printed goods may be aged before soaping, a procedure which is beneficial when printing heavy shades.

The nitrosamine method for producing insoluble azo colors is the result of the dyestuff chemist's first efforts to develop a printing process by which the naphthol could be contained in the printing paste thus dispensing with the preparing or padding operation. This method is very successful, the limitation being principally that only a comparatively narrow range of colors is available. The nitrosamine is prepared by treating a diazo salt with caustic soda, a chemical rearrangement taking place and forming a compound which may be mixed with a naphthol in an alkaline solution without reaction to form the color. Mixtures of nitrosamines and naphthols are offered on the market in paste and powder form and are extensively used for producing orange, red, and scarlet shades. The printing paste is prepared by simply dissolving the commercial nitrosamine mixture in a starch, gum-tragacanth thickener. After printing, the goods are dried and treated in a dilute acetic-acid bath, or aged in a steam atmosphere containing acetic-acid vapor or a mixture of acetic- and formic-acid vapors. In the presence of acid, the nitrosamine reverts to the diazo salt which reacts with the naphthol to form the azo color. After the color has been developed the goods are rinsed and soaped in the usual manner.

The third or diazoimino method for producing insoluble azo colors on the fiber is a comparatively recent development and is very extensively used for printing this class of colors. A wide range of shades is available and mixtures of diazoimino compounds with naphthols are offered to the textile printer in both solution and powder form under a variety of trade names.

The diazoimino process is the result of an enormous amount of research on an improvement over existing methods for applying these colors. The diazoimino compound is formed by the reaction of a diazo salt with a secondary amine, and is very stable in neutral or alkaline media but hydrolyzes very quickly in acid media regenerating the parent diazo salt. Diazoimino compounds may be mixed with a variety of naphthols, the mixtures being perfectly stable and showing no evidence of coupling, or color formation, as long as they are protected from acids. Exposure to acid vapors, however, quickly produces the azo color pigment.

The printing paste is readily prepared by dissolving the diazoimino mixture in a slightly alkaline starch, gum-tragacanth thickener. After printing, the goods are dried and then aged for from 2 to 5 minutes in an acid ager. The acid ager is similar in construction to a vat-color ager except that it is made of acid-resistant materials, and provision is made for introducing acetic acid or a mixture of acetic and formic acids. The acids vaporize at the ager temperature and the acid vapors mixed with steam induce the coupling of the intermediates, which forms the azo color. After the goods have been aged, they are rinsed and soaped in the usual manner.

It is often desirable to mix the soluble vat colors with the diazoimino colors to produce compound shades or to include the soluble vat colors in the same pattern with the azo colors, particularly when light shades are to be printed. The soluble vat colors are also developed by acid hydrolysis, accompanied however by oxidation. This is easily accomplished by adding an oxidizing agent such as sodium bichromate to the printing paste. The diazoimino color and the vat color are developed simultaneously in the acid ager.

Insoluble azo colors are only applied on white or undyed cotton fabrics and cannot be used for discharge printing. Although this is a serious limitation they are nevertheless employed for a very large percentage of the fabrics to be used for women's wear.

Basic Colors

The basic colors are extremely brilliant as a class but are inferior to vat colors and insoluble azo colors in general fastness properties. The basic colors are water-soluble salts of organic-dye bases, the acid used to form the salt being hydrochloric acid in many instances. The basic colors owe their name to the fact that the color-forming groups are contained in the basic constituent of the salt.

The basic colors are quite extensively employed on cotton, rayon, wool, and silk as they yield brilliant shades of excellent tinctorial value. On cellulosic fibers it is necessary to print these colors with a mordant in order to obtain satisfactory fastness properties. The basic colors have considerable affinity for fibers of animal origin and are often printed on wool and silk without a mordant. However, if maximum fastness is desired on these fibers a mordant should be employed.

Tannic acid is the mordant universally used with the basic colors. A printing paste is prepared containing the color, thickening agent, tannic acid, and an organic acid, usually acetic acid. After printing, the goods are dried, and steamed for approximately one hour. During the steaming, the basic color combines with the tannic acid to form a color lake which, however, is slightly soluble in water. The goods are then passed through a solution of tartar emetic which forms a double tannate of antimony and color, which is firmly fixed on the fiber and is water insoluble.

The application of the basic colors without a mordant follows the same general procedure, with the exception that tannic acid is omitted from the printing paste and the goods do not receive a tartar emetic after treatment.

The basic colors are suitable for both direct and discharge printing as they are resistant to the reducing action of sodium sulphoxylate formaldehyde. They are economical to use on account of their excellent tinctorial value and are currently employed for printing women's dress goods, draperies, and cretonnes.

Chrome Colors

The chrome colors or chrome mordant colors are acid in character and have hydroxyl or carboxyl groups in their molecules which induce reaction between the color and metallic mordants to form lakes. Chromium acetate is generally employed as the mordant for these colors although other metallic salts such as iron, aluminum, and tin salts are used for special effects. The chrome colors are of great value for printing wool although there is very little wool printing carried out in this country. Although they may be applied on all types of fibers, their most extensive use in the domestic market is for printing draperies. Many patterns are printed with chrome colors in conjunction with the nitrosamine type of insoluble azo colors. Although the chrome colors as a class are not particularly bright, the fastness properties are in general very good. They are only suitable for direct printing as the printing paste is acid in character and therefore not compatible with sodium sulphoxylate formaldehyde.

The chrome-color printing paste consists essentially of color, thickener, acetic or other organic acid, and chromium acetate. A mixture of starch and gum tragacanth is commonly employed as the thickening agent. After application of the printing paste, the printed goods are steamed for an hour or more, the insoluble metallic lake being formed during the steaming. The goods are then washed, soaped, and finished.

Acid Colors

The acid colors are so called for the reason that they are usually applied from an acid medium. These colors are employed practically exclusively on wool and silk, having no affinity for fibers of vegetable origin. A mordant is not required for their fixation as in the case of the basic and chrome colors. The acid colors consist principally of nitro compounds, azo compounds, and basic colors which have been sulphonated to impart acid characteristics.

A typical acid-color printing paste is prepared by dissolving the color in warm water, then adding sufficient British gum to form a printing paste of suitable viscosity and bringing the whole to the boiling point. The paste is then cooled to room temperature and a small quantity of acetic acid or other organic acid added. The printed goods are steamed for approximately one hour to fix the colors on the fabric. After steaming, the goods are rinsed in water, soaped lightly, and finished. Silk materials are often printed without the addition of acid to the printing paste.

Acid colors are generally employed for direct printing although a few members of the series are sufficiently resistant to reducing agents for discharge purposes. In discharge-printing pastes, the organic acid is replaced by glycerine or other hygroscopic agent. The quantity of sodium sulphoxylate formaldehyde employed should be carefully regulated, as a large excess has a deleterious effect on the acid color.

Acid colors are extensively employed in the domestic market for printing pure and tin-weighted silk for women's and children's wear.

Printing Pastes

The physical and chemical characteristics of a printing paste are influenced to a very considerable extent by the starch or gum used as the thickening agent. The choice of a thickening agent for printing a particular color depends on a large number of factors, among which may be mentioned the following:

- Chemical compatibility with color to be printed
- Depth and style of engraving of printing roll
- Color yield and brilliancy of print
- Penetration of color to the reverse side of fabric
- Sharpness of design
- Levelness and uniformity of print
- Thickening value, or viscosity per unit weight
- Working or flowing properties in the color box of the printing machine
- Solubility, or ease of removal from fabric by detergents
- Type of fiber, and fabric construction
- Cost per yard of fabric printed

Thickeners may be prepared in the color shop or purchased ready for use, a considerable number of prepared gums being available on the market which are sold under proprietary names. The commonly employed starches and gums, of which a few have been mentioned, have certain characteristics which serve as a guide to the color mixer for preparing thickeners in the color shop.

The starches such as wheat starch, cornstarch, and tapioca starch produce prints of good tinctorial value and are extensively used for printing cotton fabrics. Starch pastes containing 12 to 15 per cent of dry starch are of suitable viscosity for printing purposes. Starches adhere quite tenaciously to fabrics and it is necessary to soap the printed goods thoroughly in order to completely remove the thickener.

For printing silk and rayon fabrics it is advantageous to use very soluble gums, such as the dextrins or British gums. These products are readily soluble in cold water and may be easily removed from fabrics by a mild soaping. The dextrins and various grades of British gum are prepared from the starches by roasting or by treatment with dilute acids. British-gum pastes for printing purposes contain 40 to 50 per cent of the dry ingredients. It is common practice to mix starch and British gum together in various proportions in order to prepare thickeners for individual requirements.

Gum tragacanth is a natural gum imported from Europe and Asia Minor. This gum has very marked thickening properties, a solution containing 4 to 6 per cent of the gum being of sufficient viscosity for printing. Gum tragacanth is extensively employed with starch or British gum, as it is readily removed from the fabric and is of value in obtaining penetration of color into the fiber.

Gum Senegal and gum arabic are natural gums obtained from Africa and India. These gums are currently employed for printing silk and synthetic fibers as they yield brilliant shades and are easily removed by detergents.

Karaya gum is similar to gum Senegal and gum arabic in printing characteristics but is more economical to employ. This gum differs from the previously mentioned gums in that solutions are prepared by heating the gum with water in an autoclave under pressure.

Locust-bean gum is obtained from the seeds of the locust bean

or carob tree. The gum comes on the market in powder form and is made up into a thickener by carefully dispersing in cold water and then boiling the aqueous suspension until solution is obtained. Locust-bean gum may be employed as a substitute for gum tragacanth for printing certain colors.

The colors and gums for textile printing which have been mentioned cover only the procedures or styles which represent a considerable volume of the domestic trade. A comprehensive survey of the methods of printing in current use in this country would fill several volumes and would be of inestimable value to the textile trade. Such treatises on textile printing have been published in England and Germany and it is suggested that some public-spirited citizen prepare a similar compilation for the United States.

The future progress of textile printing calls for coordinated research on the part of the dyestuff manufacturers, the companies manufacturing gums, and textile assistants, and the textile-equipment organizations which design printing machines and auxiliary equipment. The combined efforts of all in close collaboration with the various printing plants are necessary to keep the printing industry in step with the accelerated activities in other fields of American business and industrial life.

Discussion

Coal Washing and the Baum Jig[1]

T. O. Carlisle.[2] The author has made a splendid contribution to the art of coal cleaning. Since Baum first conceived his idea, it is interesting to note that over 350 units have been installed throughout the world. The writer believes that this coal-cleaning unit has been applied more universally than any other type, and that the economic reason for the world-wide acceptance of the Baum jig is the recognized fact that over a period of 15 or 20 years, which is the average life of a jig, the cost of cleaning by this method is lower than by any other. The first cost of this jig is somewhat higher than that of other systems, but records of installations are available where the units have been in constant use for over 30 years, and in any number of instances in Great Britain where units have been operating 20 years or more. The experience of Link-Belt Company with this jig began in 1927 and the maintenance cost of installations since that has been negligible. It might be well to mention that the first Baum jig on this continent was placed in operation in Nova Scotia in 1914 and is still in daily use.

At the time the system was introduced in the United States, it was thought that the machine was fully developed to meet American conditions, but as is customary in development of new applications it was necessary to make a number of changes in the English design.

The writer believes the most radical of these came about in connection with a plant installed in 1930 wherein it was desired to install a washer to retreat the rejects from a dry-cleaning plant which contained considerable recoverable coal.

A few figures will indicate that this job was more difficult than might be imagined. Float-and-sink analysis showed that 60 per cent of the feed floated at 1.35 specific gravity, 10 per cent sank at 1.60, leaving 30 per cent between these two gravities. Moreover, 20 per cent of the feed lay between 1.35 and 1.45 gravity. To meet the clean-coal-ash requirement, it was necessary to make the separation at 1.38 specific gravity, at which point the theoretical recovery was 64 per cent.

Thus, three separate difficulties had to be overcome, namely, separation at an extremely low gravity, a feed with an unusually high reject, and a great amount of near-gravity material, that is, material whose gravity fell within 0.05 of the separating gravity. Any one of these factors presented sufficient difficulty in itself, but the combination of the three presented a real challenge. The writer was fully aware of the difficulties involved, but confidence in the Baum principle prompted us to undertake the job, even though it was pronounced impossible by some of our contemporaries.

The writer must say in all candor that the first results were not encouraging. It soon became apparent that when attempting separation at such a low gravity, with so much near-gravity material, the lightest pulsion stroke sufficient to open the bed created velocities which resulted in confusion of the mass rather than stratification.

Obviously, the solution was to produce a pulsion stroke that would open the bed to a state of complete mobility, and at the same time reduce the velocity at top stroke to obtain a longer period of settlement in relatively still water. As the author pointed out, this is the key to high efficiency in washing.

Various devices were considered and abandoned before the problem was solved by the introduction of expansion chambers. These consist of small receivers placed between the air range and the piston valves with regulating valves above and below them. The upper valves are opened just wide enough to permit a restricted flow of air. The lower valves are set as required to obtain a proper stroke. As a result, the air in the expansion chambers is partially exhausted before the pulsion stroke is completed. This imparts a quick initial lift to the bed, followed by a slower upward motion which is controlled by the restricted flow of air through the partially closed upper valves. The effect of these expansion chambers in this case was startling.

Without them, a yield of 45 per cent with a feed of 33 tons per hr have been produced. After their installation, the feed was increased to 48 tons per hr and the yield to 59 per cent. Bear in mind that the clean coal produced was for metallurgical use and the ash could not exceed 7 per cent. This was produced from a feed of the unusual characteristics described previously by the writer. Analysis of these figures shows that 5 per cent of the recoverable material at 1.38 specific gravity was still being lost, but since this contained 9 per cent ash it was obviously only the poorest quality of coal.

Since 1930, expansion chambers have been furnished on all Link-Belt washers as standard equipment, and their value in widening the field of application of this unit has been amply demonstrated. Out of 40 units in operation today, four are handling rejects from various other systems for further recovery of clean coal. One plant in particular is handling rejects during the day and screenings of an entirely different character at night with no change in adjustments required. The economies of such a setup in handling high tonnages has interesting possibilities and will, the writer believes, be more fully explored in the future.

[1] Published as paper MH-58-1, by George L. Arms, in the November, 1936, issue of the A.S.M.E. Transactions.

[2] Assistant Chief Engineer, Coal Preparation Division, Link-Belt Company, Chicago, Ill.

Coal Preparation by the Air-Sand Process[1]

David R. Mitchell.[2] Water in coal for most purposes must be considered as an impurity. Its removal in most cases presents as many technical problems as removing incombustible rock and mineral impurities. In most of the methods used for cleaning coal, water is used as the separating medium with the result that although the coal, as mined, is dry, considerable water is retained in the cleaned product after passing through the washing plant. This may amount to as much as 25 per cent for the small sizes and must be reduced by draining, filtering, or heat drying before the coal is shipped to market. Each per cent of water in coal is estimated by engineers to be equivalent to 0.7 per cent of ash in reducing the value of a given coal to the ultimate consumer. Considered from an economic standpoint, the removal of extraneous water is therefore of great importance.

One of the dreams of the coal-preparation engineer in the past has been the development of coal-cleaning processes that would clean coal in the dry state, and at the same time be as efficient

[1] Published as paper MH-58-2, by Thomas Fraser, in the November, 1936, issue of the A.S.M.E. Transactions.

[2] Assistant Professor in Mining and Metallurgical Engineering, University of Illinois, Urbana, Ill.

and of as low cost as the older methods using water. Attempts to use air were made as early as 1850 and a number of processes devised were not very successful. First successful dry plants for cleaning coal were purely an American contribution to the art of coal cleaning and began about 1919. The development of the air-sand process by the author is a distinct forward step. Data presented in this paper show the process to be efficient and to have operating costs as low or lower than many wet-washing plants.

It should also be pointed out that the cost of 7 to 8 cents a ton given by the author is probably the total plant cost, and that the actual cleaning cost is probably only about 20 per cent of this item. The other 80 per cent is made up of such items as conveying, screening, and loading which would be more or less necessary whether the coal was cleaned or not.

M. A. Kendall.[3] The air-sand process is a comparatively new development in coal cleaning, the first commercial installation being made at Cadogan, Pa., in 1930, by the Hydrotator Company. In spite of the depression, this company installed five air-sand plants for cleaning coal. The company with which the writer is associated furnished the equipment for these plants, and in 1935 took over the exclusive sale of this process and built a 200-ton per hr plant at Lundale, W. Va.

Mechanical problems involved in these plants have been taken care of by adapting conventional material-handling units. The separator boxes and the desanding screens are the only special machines which have been developed to carry out this process. Circulation of sand does not entail the serious maintenance cost originally anticipated, and it has been concluded that dry sand mixed with small percentages of coal dust can be handled without serious wear on chutes and conveyors, if abrupt turns and obstructions which might cause concentrated abrasion are avoided in plant design. For example, it has been found that No. 10 gage Salem elevator buckets, which were installed in 1930 and which have operated normally since that time, are still in good condition without replacements.

For desanding the end-products of the large separator boxes, vibrator screens much wider than were commonly used for general screening practice had to be developed. Incidentally, this has led into the development of large vibrators up to 12 ft wide suitable for handling large tonnages of screen feed.

It has been found that the most effective means of maintaining uniformly good performance in the air-sand process is the complete conditioning of the sand and the establishment of perfectly controlled circulation of the sand-cleaning medium independent of the coal-handling system. The operation of the air-sand cleaning process has been continually improved since the first plant was installed in 1930, and while a large portion of the cleaned-coal production is now prepared by systems employing water, yet the newer dry methods have important fundamental advantages that are obvious in the elimination of water. The aerated sand liquid used as the separating agent in the air-sand process functions effectively when properly controlled and obtains these advantages of dry operation.

Author's Closure

The objectives in the development of dry processes for cleaning coal must be obvious to any who are familiar with the coal business. Dry operation has two basic advantages: First, that it does not wet the coal; second, that it avoids a group of accessory plant-operating problems and costs involved in obtaining a water supply, reclaiming and clarifying it after use, preventing pollution of streams, and guarding against freezing of plant and product in winter.

The art of coal cleaning has undoubtedly reached a stage where any one of several effective processes may be adopted with confidence that an ash and sulphur reduction will be obtained that is close enough to perfect gravity separation to leave but little to wrangle over in so far as relative technological efficiencies are concerned. Choice then becomes primarily a matter of facility of operation and relative costs. I believe that the aim of the promoters of dry cleaning is to attain or approach that high degree of separating efficiency combined with the obvious advantages of dry operation. We believe that we have developed in the air-sand method a process comparing favorably in performance with the best washing methods. Since the paper under discussion was prepared, a new type of equipment for practicing the air-sand method has substantially increased the through-put per foot of separator width. This improvement, of course, has a direct relation to economy of operation. This new method of operation consists essentially in maintaining a rapid stream-flow action in the separator box, with uniform sand depth and uniform fluidity of the sand medium from end to end. This condition is conducive to rapid flow of the coal load through the box and also to rapid and effective gravity separation.

[3] Chief Engineer, Stephens-Adamson Manufacturing Company, Aurora, Ill. Mem. A.S.M.E.

The Rheolaveur Coal-Cleaning Process[1]

Byron M. Bird.[2] The author has covered in an admirable manner the importance of avoiding large fluctuations in the ash content of cleaned coal; the writer concurs entirely with his observations. If one must choose between the low ash content and a uniform ash content, he should choose the latter, because it is much the more important. The writer believes this to be equally true whether the coal is to be used in a large steam plant or in a by-product plant preparing coke for the blast furnace. In either of these instances provision can be made for the efficient utilization of coal or coke of moderately high ash content, but it is of the greatest importance after operating conditions have been established that they should not be disturbed at frequent intervals. It certainly involves no recondite reasoning to see that a large system can not be changed every few minutes or even every few hours to take care of fluctuations in the quality of the coal or the coke.

In discussing this problem the author has called attention to the rôle of the large circulating loads of the Rheolaveur in maintaining a uniform cleaned coal. But he has not mentioned another item that the writer regards as of greater importance in most plants than the circulating load, and that is the bed of material that commonly accumulates on the bottoms of the launders. Fig. 1 of the paper shows all of the material in the launder as moving along the bottom. The writer's observation has been that this condition is relatively uncommon. Between refuse draws most launders have what, for want of a better term, the writer shall call a "static" bed. This forms the effective bottom of the launder, and the stratification occurring in the stream takes place on top of this bed. As long as the character and the tonnage of the feed and the horizontal velocities of the water are unchanged, this bed remains fixed.

Perhaps the writer can clarify this point somewhat by considering for a moment a simple launder with no refuse pockets in the bottom. Let us start with a condition in which the launder is

[1] Published as paper MH-58-3, by John Griffen, in the November, 1936, issue of the A.S.M.E. Transactions.

[2] Chief Concentration Engineer, Battelle Memorial Institute, Columbus, Ohio.

sloped sufficiently for all of the materials to move along with the water, that is, the condition shown in Fig. 1 of the paper, and let us gradually decrease the inclination. After a time a point will be reached at which some materials deposit from the moving stream. Once equilibrium is established between the materials transported by the stream and those comprising the static bed, the top of this bed will take a definite slope. The angle between the top of this bed and the horizontal, when neither erosion nor deposition takes place, has been termed by hydraulic engineers "competent slope." As long as conditions remain constant, this bed is fixed. If we introduce refuse pockets into the bottom of the launder and start withdrawing portions of the bed, a selective deposition from the flowing stream takes place over each pocket in which materials mainly of the highest specific gravity are deposited. If a given pocket is operated to draw a comparatively high tonnage in relation to its area, a high percentage of coal may be deposited from the stream along with the refuse and the competent slope of the bed ahead of the given pocket may undergo some change. However, the most important factors governing the static bed are the character and tonnage of the feed to the launder and the horizontal velocity of the water.

In Rheolaveur plants it is common practice to operate the launders at angles to the horizontal less than competent slope so that static beds form along the bottom. The beds so formed have a very important effect upon the capacity of the system to absorb variations in the feed. If the tonnage is increased, all other conditions remaining the same, the static beds become steeper and, hence, thicker. In this manner the launder can absorb a rather marked fluctuation in the tonnage without readjustment. Similarly, if there is a peak point during which the percentage of refuse in the feed is high, the static bed will build up between the refuse draws in the establishment of a new competent slope. This change within the launder is sufficient to take care of a rather pronounced fluctuation in the character of the feed. If the percentage of refuse drops unusually low, the reverse happens and the stream erodes a part of the static bed. Thus, the static bed is an important factor in stabilizing a Rheolaveur system.

David R. Mitchell.[3] It is not only desirable that coal should be low in ash, sulphur, and other impurities, but it should also be uniform as to the amount of impurities present in successive shipments. Steam plants can be designed and constructed to burn efficiently coal containing high percentages of ash. However, if consignments vary widely in ash content, the cost of producing steam rises because the morale of the personnel of the boiler room tends to be lowered, and it becomes increasingly hard to obtain efficient operation.

Very little information is available in the technical literature on the uniformity of the product from coal-preparation plants. Analyses of samples taken in the mine are not representative of the quality of the coal shipped and purchasing agents are continually confronted with the problem of determining the quality of the coal as it arrives at the consuming plant. Also, they are faced with the problem of evaluating coal upon which bids are received. Will the coal from the various mining companies fluctuate widely from the bid analysis or will the purchaser be able to say that it is probable that certain coals will show little variation from the bid analysis because of the preparation given the coal at the mine? Certainly of two coals of the same ash content, the one that fluctuates the least from the average will be the best to buy. Cognizance must be taken of the fact that in one case variations in shipments will show variations in ash content of the average plus or minus 5 per cent; while coal from the same district and coal bed may only show variations of the order of plus or minus 0.5 per cent because of the preparation given the coal. A boiler plant cannot be expected to operate at maximum efficiency when the coal furnished one day analyzes 15 per cent ash and the next day only 5 per cent.

The writer believes that the author has made a real contribution to the literature on coal cleaning by showing from actual preparation-plant studies the remarkable uniformity of cleaned coal over uncleaned, and specifically on coal cleaned by the Rheolaveur process. This should be of immense value to purchasing agents; technical literature contains very little information of this kind. In fact, for the great number of processes used for cleaning coal very little reliable operating information is available at all, except possibly those plants in operation in the Pennsylvania anthracite-coal fields.

The first Rheolaveur plant in the United States was erected in 1926. This and succeeding plants were so successful in cleaning coal to a low and uniform impurity content at low cost, that a consistent and steady growth in the use of this process has taken place.

The author gives the impression that the skewness of ash-fusion analysis is always on the minus side. Is it not true that this condition probably exists for coal in which the cleaned coal is higher in ash - fusion temperature than the uncleaned? Also to express variability he uses graphic methods and mathematical calculations, presumably the Standard or Peter's equation for probable error. Modern statisticians seem to favor expressing results in terms of standard deviation rather than probable error. Is there any particular advantage in a study of coal variability if results are expressed in terms of probable error rather than standard deviation?

Also, it might be mentioned that many coal-mining companies take enough penalties, on a few shipments of raw coal that run to high ash, to more than pay for cleaning the entire output. Further, it might be added that if the preparation plant has poor facilities for screening, conveying, mixing, loading, and otherwise handling the coal after it is cleaned, much of the desirable features of cleaning may be negative. It is just as desirable that coal arrive at the consuming plant uniform as to size and size distribution as that it be of uniform and low impurity content.

J. W. Stewart.[4] In his description of the performance of the Rheolaveur plant *A*, cleaning coal from an Illinois strip mine, the author refers to Table 2 of the paper and points out, with reference to the duff size ($^1/_4$-in. to 0-in. coal sizes) that: "It is interesting to note that this smallest size is cleaned to the lowest average ash content. . . ." This is quite true, but it might be questioned whether there was any noticeable improvement in the finer sizes of this range, say finer than 65 mesh. Seyler[5] says that in his experience, when using the Rheolaveur process in cleaning coal from the Pittsburgh seam of Pennsylvania, he has failed to find noticeable improvements effected upon coal finer than 100 mesh in size. Walle and Woody[6] report that in washing a cretaceous coal from the Trinidad region of Colorado, they found that it was unprofitable to attempt cleaning the portion finer than 48 mesh.

(The discussion which follows was presented jointly with the discussion of the paper by George L. Arms.[7])

[3] Assistant Professor of Mining and Metallurgical Engineering, University of Illinois, Urbana, Ill.

[4] Assistant Professor of Mining Engineering, Pennsylvania State College, State College, Pa.

[5] "Washing Coal for Coking Purposes at Clairton By-Product Coke Works," by H. W. Seyler, *Coal Age*, vol. 38, June, 1933, p. 192.

[6] "Metallurgical Coal Washing Plant of American Smelting and Refining Company at Cokedale, Colorado," by M. Walle and G. V. Woody, *Mining Congress Journal*, vol. 13, March, 1927, p. 198.

[7] "Coal Washing and the Baum Jig," by George L. Arms, Trans. A.S.M.E., vol. 58, November, 1936, paper MH-58-1, pp. 697–699.

Henry F. Hebley.[8] Papers on the subject of coal preparation are most timely because the subject is of great importance. There should be closer cooperation between the engineers who actually use the fuel and the engineers who prepare it for use. Although tangible results have been attained through the cooperation of the A.S.M.E.-A.I.M.E. Joint Committee on Fuel Evaluation, the subject should be made more familiar to all members of the societies.

Coal should be prepared from the utilization point of view. The requirements for metallurgical coal may not be the same as for steam generation, gas generation, or the ceramic industry; and it is these requirements which should be stated by the mechanical engineer in order that the most suitable preparation will be attained.

Coal preparation, in the larger sense, includes cleaning, sizing. dust and moisture removal, and a host of auxiliaries incident to the main functions. For the most part, the separation of refuse material from coal is dependent on density differences between the materials, although such properties as the difference in friction coefficients, and the difference in resiliency form the bases of certain coal-cleaning equipment. Water is the medium in many systems, chief of which are the launder, jigging, and concentrating tables. The use of chemicals or powdered solids in conjunction with water to increase the specific gravity yields a second class of machines which can properly be called "density liquid" machines. Dry-cleaning equipment is available, using air as the medium of separation, and it too can be arranged in the form of launders, jigs, and concentrating tables.

No coal-cleaning apparatus yields coal with no ash, but the ash content can be materially reduced. More important, however, than extremely low ash, is the uniformity of product turned out by the cleaning plant. Remembering that coal in its natural state is a heterogeneous mixture of "coalified" wood which has been contaminated by dirt and clay, and that is further contaminated by sand, clay, shale, and rock, during mining, any commercial treatment to which it is subjected which will improve the coal will still leave some variation between the different shipments of fuel. It is this lack of uniformity, or variation, which coal-preparation engineers are making every effort to overcome.

Using the ash content of coal as the measure of a coal's uniformity, it is common procedure to express results in the "average ash." Such a measure, as we well know, gives a very incomplete picture of the uniformity of a plant's production. It is gratifying to have the author's data presented using the statistical method for reporting the measure of a plant's production. The author's curves show clearly the range, the central tendency, and the probable error; therefore, standard deviation can be obtained from his data. A study of the curves submitted shows how great is the improvement in the uniformity of the washed-coal product.

A comparison of the uniformity of coals by specific gravities would be of great interest if available. For instance, if one were to take raw-coal samples and subject them to a density separation at 1.55 specific gravity, and also take washed-coal samples and subject them to separation at the same density, would the ash of the raw coal floating be less uniform than the ash of the floating washed coal?

The data presented in this paper bring up the question of correct sampling, and the interpretation of results. Until recently no great advance had been made in coal sampling since Bailey published his results.[9] Now, however, Morrow and Procter,[10] Kassel and Guy,[11] Grunnell and Dunningham,[12] and Pearson[13] have all published excellent treatises on the subject. Articles on sampling and plant control have appeared in *Mechanical Engineering*, and the A.S.T.M. Manual on the Presentation of Data gives an excellent guide to those who wish to use this method. It is to be hoped that greater application will be made of this extremely useful tool for measuring the output of a plant.

Coal which is laminated with carbonaceous shale and having a specific gravity between 1.30 and 1.60 (often called "bone coal" or "middlings") is difficult to separate from higher ash refuse because of the nearness of the respective gravities. When a large percentage of such material is present in a coal seam, the cleaning process is apt to be involved. Generally it calls for skimming off the clean coal and pure refuse, and then taking the difficult material and recirculating it for further treatment. Crushing the middlings may be adopted just prior to recirculating in order to release the laminated coal. Both the launder system and the Baum jig lend themselves to large capacity and difficult washing. The latter machine has been well described by Arms,[7] and it represents the latest unit based on the Baum principle to be developed in this country.

The dynamics of a jig washer are rather involved and there is not much information available on the subject. The relation of the acceleration of the water, on both the pulsating and suction stroke, to the air pressure, the influence of the "cushioning" effect of the air on the "back suction" during the suction stroke, the speed of flow of the material from inlet to outlet, and the length of jig box travel for separation, are all matters of importance, and further contributions covering these phases would be most helpful. (The following discussion was submitted jointly with the discussion of the paper by George L. Arms[7] and the paper by Thomas Fraser.[14])

C. W. Hunter.[15] The writer gathers that these papers[1,7,14] are intended to constitute a symposium on coal-washing methods as practiced in the United States today. Therefore, it is surprising that no mention has been made of one of the major systems which is today cleaning nearly one half of the anthracite output, as well as being in operation in many plants in this country and abroad for cleaning bituminous coal, and which operates on a principle entirely different from those described. The writer refers to the Chance sand-flotation process, invented by an American engineer and developed under American conditions.

The Chance sand-flotation process for the separation of coal and refuse is a straight flotation system. The flotation medium is secured by the suspension of sand in water and its specific gravity is proportional to the relative amounts of sand and water present. In practice, this is varied to suit the coal, usually within the range of 1.38 to 1.75 specific gravity. The coal sub-

[8] Engineer in Charge of Coal Preparation, Commercial Testing & Engineering Company, Chicago, Ill. Mem. A.S.M.E.

[9] "Accuracy in Sampling Coal," by E. G. Bailey, *Journal of Industrial and Engineering Chemistry*, vol. 1, March, 1909, p. 161.

[10] "Variables in Coal Sampling," by J. B. Morrow and C. P. Procter, Technical Publication No. 645-F-67, American Institute of Mining and Metallurgical Engineers, September, 1935.

[11] "Determining the Correct Weight of Sample in Coal Sampling," by L. S. Kassel and T. W. Guy, *Industrial and Engineering Chemistry*, Analytical edition, vol. 7, no. 2, 1935.

[12] "Report on the Sampling of Small Fuel Up to 3 In. Embodying Some General Principles of Sampling," by E. S. Grunnell and A. C. Dunningham, Report No. 403, British Engineering Standards Association, December, 1930.

[13] "Application of Statistical Methods to Industrial Standardization and Quality Control," by E. S. Pearson, British Standards Institution, London, publication no. 600, November, 1935.

[14] "Coal Preparation by the Air-Sand Process," by Thomas Fraser, Trans. A.S.M.E., vol. 58, November, 1936, paper MH-58-2, pp. 701–703.

[15] Vice-President, United Engineers & Constructors, Inc., Philadelphia, Pa. Mem. A.S.M.E.

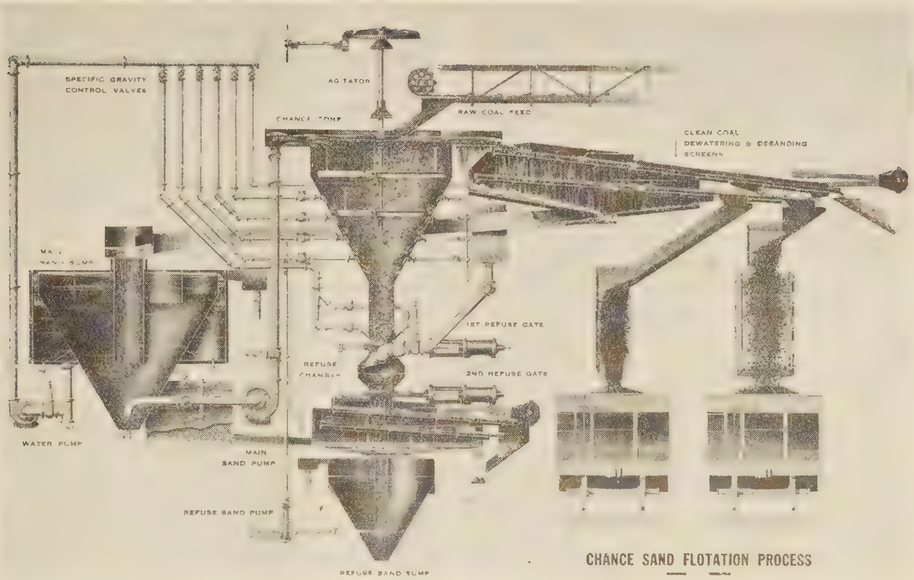

Fig. 1 Sectional View of the Chance Sand-Flotation Process

stance, which is lighter than the selected specific gravity of the medium, floats irrespective of its size or shape; the refuse which is heavier, sinks. The Chance process is, in fact, a plant scale application of the float-and-sink test (the standard laboratory method of testing coal separation), in which the costly heavy chemicals used in the laboratory to secure high specific gravities are replaced by sand and water, about the cheapest raw materials available. So accurately does the system separate the coal and refuse that records of active plants frequently show long periods of operation where the laboratory float-and-sink tests on the cleaned coal, made at the operating gravity, show an average of less than 1 per cent of sink material; and float-and-sink tests at the same gravity on the refuse average less than 1.5 per cent of float material. Such a near approach to the theoretical is rarely found in any practical engineering process in any industry.

The heart of the process is the Chance cone, of welded steel construction, in which a flotation medium of sand suspended in water is maintained at the specific gravity at which it is desired to separate the refuse from the coal. The sand is kept in suspension partly by mechanical agitation and partly by gently upward-flowing currents of water. The raw coal is introduced to the top of the cone at the surface of the fluid mass. The good coal, being of lower specific gravity, floats smoothly around the cone for about three fourths of its periphery and is discharged onto the desanding and dewatering screen, from which it passes to sizing screens and loading booms.

A layer of sand-free water is provided on top of the fluid mass, through which the floating coal has to pass when it is discharged to the desanding and dewatering screen, and any particles of sand that may adhere to the coal are washed off. As a precautionary measure, additional clear water is sprayed onto the coal while it passes over the screen. The water carrying the sand which is washed off the coal passes through a flume to the main sand sump—another steel cone set at a lower elevation.

All material in the raw coal feed, which is heavier than the fluid mass, sinks through the mass to the bottom of the Chance cone and passes through an automatically operated gate into the refuse chamber. From the refuse chamber it is intermittently dropped through a second automatically operated gate onto the refuse desanding and dewatering screen and from there is conveyed to refuse bin or cars. Such sand as may adhere to the refuse is washed off by means of the water discharged with it from the cone and by clear-water sprays over the screen. The sand and water from the refuse desanding screen pass into the refuse sand sump, a smaller steel cone, where it is picked up by a centrifugal sand pump and deposited in the flume leading from the coal desanding and dewatering screen to the main sand sump.

The sand and water enter the main sand sump through a pipe extending well down into the sump. The clear water rises to the surface and overflows into the clear-water reservoir compartment, which surrounds the upper part of the sump while the sand settles to the bottom and is elevated as needed by a centrifugal sand pump to the top of the cone. The clear water is pumped from the clear-water reservoir and is delivered to the water-agitation manifolds leading to the cone and to the sprays.

When required, make-up sand can be automatically admitted to the sand sump. The required specific gravity is maintained in the cone by regulating, either manually or automatically, the quantity of clear water admitted through the water-agitation manifolds to the cone. A cross section of a typical Chance bituminous installation is shown in Fig. 1 of this discussion.

The outstanding feature of the Chance process is the sharp cut that it makes between coal and refuse, which results in the lowest ash and sulphur content in the cleaned coal and minimum refuse losses; thus, the process is especially adapted to the production of metallurgical and by-product coals.

Author's Closure

The author appreciates Mr. Bird's calling attention to the regulating effect of the static beds in Rheolaveur launders. They undoubtedly have such a regulating effect, but the author's experience does not suggest that they are as effective as the circulating load which is termed "regulating material." Without a doubt, these regulating features of the Rheolaveur process minimize the need of operator supervision and adjustment, and insure a high degree of uniformity in the quality of clean coal and refuse.

Referring to D. R. Mitchell's comments, it is probably true that the statement that "the skewness of ash-fusion temperatures is on the minus side" should be limited to coals which show a higher ash-fusion temperature when refuse is removed. Although such

a result is not universal, it is undoubtedly found with the majority of coals mined in the United States, even if the improvement is only slight. Regarding the use of probable error, rather than the standard deviation, the former was used because it has been used generally in the work on coal variability to which reference was made in the paper.[1] Further, the author thought that probable error might be more easily understood by those in the coal industry, that is, both producer and purchaser.

J. W. Stewart's comments on the cleaning of coal finer than 65 mesh are true and apply with equal or greater force to all other types of equipment dependent upon specific-gravity separation. Such particles are less than 0.01 in. diameter, and differences in specific gravity become ineffective in overcoming the conditions inherent in their large surface area relative to their mass. Commercially this point does not seem important since this size usually represents but 1 per cent of the mine output. However, the point to be emphasized is that $^1/_4$-in. to 48-mesh coal had been cleaned so well by the Rheolaveur process that when shipped it showed the lowest average ash content and variability. This is a condition which will not be found generally in the product from other cleaning equipment and certainly not when a wide-size-range coal, such as 4-in. to 48-mesh sizes, is handled in one unit. Under such a condition, cleaning efficiency will usually drop very low on coal smaller than $^1/_8$ in., which is sufficient in quantity to have definite commercial significance.

H. F. Hebley raises an interesting point regarding the relative uniformity of washed coal and the raw coal floating at some gravity like 1.55. The following will answer this question:

Plant *F* is operating on coal from the Pittsburgh seam and consists of a coarse-coal Rheo unit similar to that shown in Fig. 2 of the paper, but to which was added a crusher, a third final rewash launder, and a fine-coal Rheo unit similar to that shown in Fig. 4 of the paper. The refuse from the unit similar to that shown in Fig. 2 is crushed to pass $1^1/_2$-in. screens and is recleaned in the final rewash launder. All the $^5/_{16}$-in. to 0-in. material is given a final cleaning in the fine-coal Rheo unit. All the slurry is recovered by a Dorr thickener and filters, and is then added to the washed coal.

The run-of-mine coal is placed in a 5000-ton blending bin, divided into compartments, from which it is drawn at the rate of 650 to 700 tons per hr. It is then crushed to pass 4-in. screens and is delivered to the cleaning plant. The blending bin materially reduces the variability of the raw coal fed to the washing plant. The variability data are given in Table 1 of this discussion.

TABLE 1 VARIABILITY OF ASH AND SULPHUR CONTENT OF 4-IN. TO 0-IN. RAW AND CLEANED COAL FROM THE PITTSBURGH SEAM, PLANT *F*

	Raw coal		
	Total	Floating at 1.55 specific gravity	Cleaned coal
Number of days sampled	351	116	353
Number of increments per day	16	16	32
Weight of each increment, lb	375	375	375
Ash, dry basis, per cent:			
Average	9.2000	6.5000	6.8000
Probable error	0.3890	0.1870	0.1570
Sulphur, dry basis, per cent:			
Average	1.6165	1.2100	1.2570
Probable error	0.0818	0.0314	0.0440

The Effect of Installation on the Coefficients of Venturi Meters[1]

Robert W. Angus.[2] It is now almost universal practice to use the venturi meter to measure the discharge from pumps. In many acceptance tests conducted by the writer, the venturi meter, and the coefficients already reported by Professor Pardoe, were used exclusively for measuring pump discharges.

In all the installations examined by the writer the tubes have been placed in a straight line of pipe ten or twelve diameters from any elbows or changes of section, and as he read the paper he wondered why anyone would expect satisfactory results from many of the settings shown in the paper. The writer made a careful series of experiments to determine the errors possible in pressure measurement and believes that part of the error found in the meter is due to an incorrect pressure indication on the upstream side.

The tubes used by Professor Pardoe had the pressure openings properly made and were not near lumps or projections in the pipe, as the latter have been found to give pressures differing from the true pressures by the velocity head in the pipe.

The theory indicated by the author assumes, as shown in Fig. 3 of the paper, an elliptic velocity distribution. This is open to some question. He has also introduced a term α for the purpose of correcting the velocity head for the irregular velocity distribution. Undoubtedly, some correction is necessary because the velocity used in the Bernoulli equation is the quotient of the discharge divided by the area, whereas the velocity head depends on the square of the velocity, and hence on the average of the velocity squared of the several particles. The writer, however, believes that a more helpful method is to throw all of these irregularities into the resistance head and make it account for the inaccuracy of the Bernoulli equation. This conception is simple and makes the solution of problems easy.

To adopt this method with the author's equations, starting with Equation [2] of the paper

$$Z_1 + P_1 + \alpha \frac{V_1^2}{2g} = Z_2 + P_2 + \frac{V_2^2}{2g} + k\frac{V_2^2}{2g}$$

one can write $\alpha = 1 + \beta$ and $\beta(V_1^2/2g) = \beta(d_2/d_1)^4(V_2^2/2g)$ and Equation [2] of the paper becomes

$$Z_1 + P_1 + \frac{V_1^2}{2g} = Z_2 + P_2 + \frac{V_2^2}{2g} + \frac{V_2^2}{2g}\left[k - \beta\left(\frac{d_2}{d_1}\right)^4\right] \quad \text{..[1]}$$

and Equation [14] of the paper would then be written

$$V_2 = \frac{1}{\sqrt{\left[1 - \left(\frac{d_2}{d_1}\right)^4 + \left\{k - \beta\left(\frac{d_2}{d_1}\right)^4\right\}\right]}} \sqrt{[2g\,(P_1 - P_2)]} \quad \text{..[2]}$$

or treating the term $[k - \beta(d_2/d_1)^4]$ as the part commonly included in resistance and writing it as a coefficient of resistance c_r there results

$$V_2 = \frac{1}{\sqrt{[1 - (d_2/d_1)^4 + c_r]}} \sqrt{[2g(P_1 - P_2)]} \quad \text{......[3]}$$

Experiments on the tubes may then be used to obtain the coefficient c_r for the reducing cone, and these coefficients may be used in computing the value of the coefficient C given in Equation [1] of the paper. This method applied to the 11.956 × 8.2285-in. meter with a coefficient C of 0.982, as given in Table 1 of the paper, gives $c_r = 0.0287$. For the 7.81 × 5.0046-in. meter, $c_r = 0.0221$. On the 8.06 × 3.3759-in. meter, the resulting $c_r = 0.0196$ which also appears to be consistent. For a value of $c_r = 0.022$, the resulting value of the coefficient C is 0.9844, differing by only 0.56 per cent from the measured value.

Ronald B. Smith.[3] The author's experiments on the in-

[1] Published as paper HYD-58-6, by W. S. Pardoe, in the November 1936, issue of the A.S.M.E. Transactions.

[2] Professor of Mechanical Engineering, University of Toronto, Toronto, Canada. Mem. A.S.M.E.

[3] Turbine Engineering Department, Westinghouse Electric & Manufacturing Company, South Philadelphia Works, Philadelphia, Pa. Jun. A.S.M.E.

fluence of the velocity profile at the entrance to a venturi meter on the discharge coefficient of the tube offers the possibility of interpreting the effect of roughness in the approach pipe. For some years it has been recognized that upstream roughness increased the discharge coefficient of a primary element. With perfectly smooth pipes, the velocity profile in the region of turbulent flow is rectilinear with the exception of the boundary layer, and corresponds essentially to the condition the author prefers to describe as "shooting flow." Increased roughness tends, by virtue of the added shearing stresses, to peak the velocity profile. Under normal conditions the roughness effect would hardly ever produce a relative profile change as marked as the comparison between Figs. 11 and 9 of the paper. Qualitatively, then, the effect of approach profile, summarized in Fig. 4 of the paper, may be interpreted as marking the maximum effect of upstream roughness on the discharge coefficient of a venturi tube, and in view of their similarity, a nozzle as well. In this light the results are in accord with the experimental roughness corrections for nozzles specified by the V.D.I. Regeln, as indicated in Table 1 of this discussion.

TABLE 1 COMPARISON OF V.D.I. AND PARDOE CORRECTION FACTORS

Diameter ratio	0.420	0.641	0.69
Maximum correction given by V.D.I. for 2-in. pipe, per cent	0	0.350	0.60
Maximum correction for profile determined by Pardoe, per cent	0	0.500	0.70

The influence of helical flow on the discharge through primary elements has always been viewed with seriousness, yet the author's experiments form, as far as the writer is aware, the first quantitative evidence of the effect. For that reason an analytical treatment may be of interest.

Let us consider the effect in a pipe of a pure rotation superimposed on a linear velocity. If the subscripts 1 and 2 correspond to the upstream and the throat positions, respectively, and the subscripts w and z denote the tangential and the axial components, respectively, the absolute velocity c may be represented as

$$c^2 = c_w^2 + c_z^2 \qquad [4]$$

The strength of the vortex remains constant, and its behavior is characterized by the relation

$$c_w r = \Gamma/2\pi = \text{const} \qquad [5]$$

where r is the radius and Γ the strength of the vortex.

The energy relation may be written with sufficient accuracy in the form

$$\frac{P_1}{\gamma} + \frac{c_{1_z}^2}{2g} + \frac{c_{1_w}^2}{2g} = \frac{P_2}{\gamma} + \frac{c_{2_z}^2}{2g} + \frac{c_{2_w}^2}{2g} \qquad [6]$$

and the continuity relation as

$$c_{1_z} r_1^2 = c_{2_z} r_2^2 \qquad [7]$$

Denoting the area ratio of the venturi meter by $m = (r_2/r_1)^2$ and noting that

$$\left.\begin{aligned} c_{2_w}^2 &= (1/m)c_{1_w}^2 \\ c_{1_w} &= c_{1_z}\cot\alpha \end{aligned}\right\} \qquad [8]$$

where α is the vortex angle noted in Fig. 44 of the paper, we find that

$$\frac{P_1 - P_2}{\gamma} = \frac{c_{2_z}^2}{2g}(1 - m^2)\left(1 + \frac{m}{1+m}\cot^2\alpha\right) \qquad [9]$$

For an axial flow without rotation the pressure drop is

$$\frac{P_{1_0} - P_{2_0}}{\gamma} = \frac{c_{2_z}^2}{2g}(1 - m^2) \qquad [10]$$

so that the effect of a superimposed vortex is to produce an apparent increase in the measured differential between the upstream and the downstream pressure connections.

The effect on the discharge coefficients may be represented by rearranging Equations [9] and [10] of this discussion in the form

$$\frac{\varphi'}{\varphi} = \frac{1}{\sqrt{(1 + [m/(1+m)]\cot^2\alpha)}} \qquad [11]$$

where φ' is the discharge coefficient with a helical flow and φ is the coefficient without rotation. For slight rotation or small m, the change is approximately

$$\frac{\varphi' - \varphi}{\varphi} = -\frac{m}{1+m}\,\tfrac{1}{2}\tan^2\alpha \qquad [12]$$

In Fig. 1 of this discussion are plotted the results shown in Fig. 44 of the paper as a function of the variable $1/\sqrt{[1 + (m/1 + m)\cot^2\alpha]}$. The agreement is surprisingly good.

At higher axial velocities the experimental coefficients apparently fall in a random fashion; however, this is not entirely unexpected because the strength of the vortex increases directly as the axial velocity. With high-strength vortices, the discontinuity near the center as indicated by the relation $c_w r$ = constant may be quite severe.

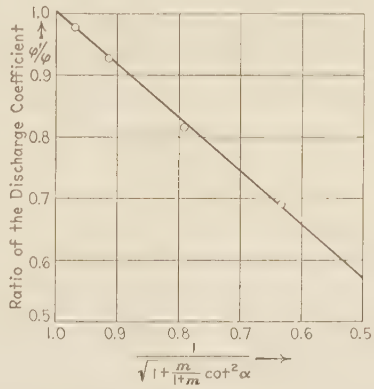

FIG. 1 EFFECT OF ROTATION ON THE DISCHARGE COEFFICIENT OF A VENTURI METER
(From Professor Pardoe's tests.[1])

In conclusion, Equation [11] of this discussion leads to an interesting result: The effect of helical flow like other flow disturbances is less severe with small area ratios. Perhaps the author can verify the theory.

ED S. SMITH, JR.[4] It is generally helpful to consider present works in the light of those preceding them, and therefore the writer will refer to previously published notes on the subject under discussion, some of which were published by the writer.

The coefficient for the venturi meter was given in 1929 as

$$C = \sqrt{\left[\frac{R^4 - 1}{\alpha_2 R^4 - \alpha_1 b}\right]}$$

which may be rewritten as

$$C = \sqrt{\left[\frac{1 - (d_2/d_1)^4}{\alpha_2 - \alpha_1(d_2/d_1)^4 + b(d_2/d_1)^4}\right]}$$

from which Equation [4] of the paper is derived if the assumption is made that there is uniform velocity distribution at the throat,

[4] Hydraulic Engineer, C. J. Tagliabue Manufacturing Company, Brooklyn, N. Y. Mem. A.S.M.E.

i.e., if α_2 is unity. The author's unconventional use of the term "shooting flow" is objected to by the writer as misleading in view of the accepted use of the term in a different sense with open-channel flow. Would not the term "uniformly distributed flow" be more generally acceptable?

Test results[5] published by the National Bureau of Standards in connection with air and gas meters bring out the fact that the velocity at the throat falls off near the walls in a manner that is intimately related to the value of the coefficient. With large (d_2/d_1)-ratio tubes the effect is far from negligible; in fact, with ratios over 0.75 the walls of the throat hardly affect the main flow enough to give a substantially uniform velocity distribution at the throat.

In view of the present state of the art it seems unnecessary to resort to the assumption that the velocity distribution be treated as an ellipse. In this connection, J. F. Downie Smith[6] and the writer[7] have both given a more precise method where the velocity distribution is known.

Like F. zur Nedden's assumption[8] that variation of roughness does not appreciably affect eddy losses, the author's assumption that variation of inlet velocity distribution does not appreciably affect the value of the friction factor k is of much use only as long as the section is contracting, which is where the author used it. In other words, this assumption should not be extended offhand to cover losses for the entire tube[9] since a rougher inlet cone causes the flow to follow a more abruptly expanding discharge cone than with a smoother inlet. The venturi meter is so efficient that its loss can properly be determined only between points of normal velocity distribution upstream and downstream of the tube. An abnormal velocity distribution upstream of the tube is likely to affect the loss seriously. Fig. 44 of the paper agrees with other data known to the writer, both as to coefficient and loss, and differs with the last sentence on page 70 of the A.S.M.E. Fluid Meter Report, Part 1, third edition, 1931, which states that it is known that a strong initial twist may give a discharge coefficient greater than unity. However, the losses in a relatively short expanding cone may be decreased considerably by giving the flow a twist that aids it to fill the diverging cone.

Although the accuracy of the paper is high, the author would increase its usefulness if he would make a statement of (1) the value of the gravitational acceleration g that he actually used in his computations, (2) any effects of cavitation on the coefficients presented, and (3) the roughness of the inlet and throat portions of the tubes tested and described as "venturi tubes of the Herschel type."

R. E. Sprenkle.[10] The results obtained from the author's experiments are quite interesting and worth-while in the application of venturi tubes to commercial metering problems. More information along similar lines not only about venturi meters, but every type of primary element in common use is sorely needed, and it is hoped more experimenters will contribute as effectively to the public fund of knowledge about installation conditions as has the author.

Along this line, the writer wishes to call attention to a report issued some two years ago, by the Hydraulic Institute of Munich, Germany, entitled "Influence of Bends Preceding Venturi Tubes," by H. Mueller, in which valuable data also are presented on installation conditions. Mueller used 4-in. venturi meters of diameter ratios of approximately 25 per cent, 36 per cent, and 50 per cent. Calibrations were first made with 56 pipe diameters preceding the venturi so as to obtain the normal coefficient, just as the author has done in his work. Then the venturi in each case was moved up to immediately precede a 90-deg elbow and further tests were made. This installation simulated that illustrated in Fig. 24 of the paper. After tests had been made with this setup, the installation was again changed so as to have two 90-deg elbows at right angles to each other immediately ahead of the venturi, this being comparable to Fig. 32 of the paper.

The results of all these tests showed that even the double-bend combination did not produce errors in the flow measurement greater than 0.25 per cent, the resulting coefficients also being low. Naturally the single elbow produced even smaller errors.

Tests were also made with two diameters between both the single and double bends and the venturi inlet, with so small an error in each case as to be scarcely discernible.

While these German experiments show somewhat smaller installation errors than Professor Pardoe's, it must be remembered the German design of venturi tube is not quite the same as the American, in that the inlet section more nearly resembles a well-rounded flow nozzle in contour, which may in itself account for all the differences in results. At best, Professor Pardoe's data and that of Mueller's are sufficiently in agreement to warn the prospective user to stay as far away as possible from bends on the inlet side of a venturi meter if highly accurate results are to be expected.

The Rounded-Entrance Venturi Meter

Alfred E. Sorenson.[11] The purpose of the writer's discussion is to show the effects of a different type of entrance section to a venturi meter, rather than the effects of installation. Fig. 2 of this discussion shows a 5 × 2½-in. venturi meter made from brass castings and installed in the 5-in. discharge line of a 700-gpm pump. The venturi is installed 17 diameters downstream from the nearest elbow, but, on the downstream side, a flanged cross is bolted directly to the meter so that the quantity of water flowing can be measured for three separate experiments. The converging section was machined to a template by hand tools and is made similar in proportions to a rounded-entrance orifice. Excellent design data for such nozzles have been given by Buckland.[12] The diverging section is of the usual type with a 4-deg half angle.

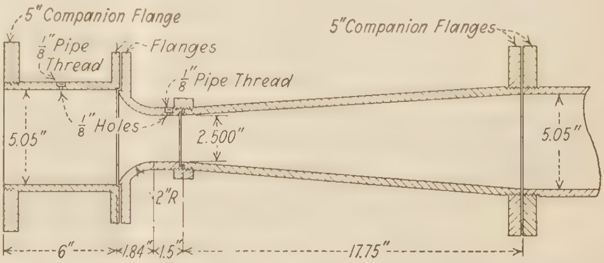

Fig. 2 Cross Section of the Rounded-Entrance Venturi Meter

[5] "Discharge Coefficients of Square-Edged Orifices for Measuring the Flow of Air," by H. S. Bean, E. Buckingham, and P. S. Murphy, National Bureau of Standards *Journal of Research*, vol. 2, March, 1929.

[6] Discussion of "Air Flow in Fan-Discharge Ducts," by L. S. Marks, A.S.M.E. Transactions, vol. 57, 1935, p. 347.

[7] Ibid., p. 348.

[8] "Induced Currents of Fluids," by F. zur Nedden, Trans. A.S. C.E., vol. 80, 1916, p. 844.

[9] "Hydraulics," by R. L. Daugherty, McGraw-Hill Book Co., Inc., New York, N. Y. Third edition, p. 134.

[10] Mechanical Engineer, Bailey Meter Company, Cleveland, Ohio. Mem. A.S.M.E.

[11] Assistant Professor of Mechanical Engineering, Princeton University, Princeton, N. J. Mem. A.S.M.E.

[12] "Fluid-Meter Nozzles," by B. O. Buckland, Trans. A.S.M.E., vol. 56, November, 1934, paper FSP-56-14, pp. 827–832.

There are two main advantages in using this type of entrance section: (1) The flow pattern at the throat should be more uniform than in the straight converging type and much less sensitive to the surface condition of the entrance section; and (2) there is a great saving in space required for the installation of the meter as well as the amount of metal required in making it. The obvious disadvantage is the care required in machining a section of this shape, but this is not serious.

The pressures at the entrance and throat were measured by the writer on differential gages very similar to those used by the author in his experiments. Because the writer had no weighing tanks large enough for the quantity of water discharged by this meter, he measured the water volumetrically in a tank 14 ft long, 4 ft wide, and $2^1/_2$ ft deep. This tank was previously calibrated by letting in water from a weighing tank above and noting the rise in level on a hook gage inside the tank.

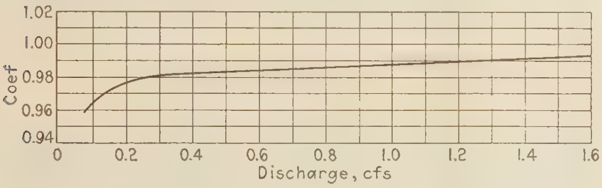

Fig. 3 Meter Coefficients Versus Discharge

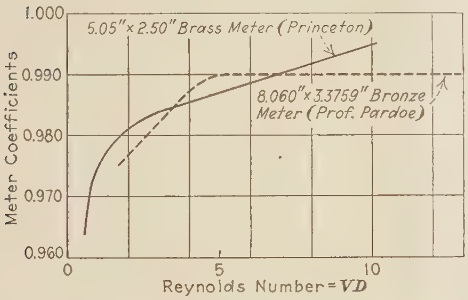

Fig. 4 Meter Coefficients Versus Reynolds' Numbers

Fig. 3 of this discussion shows a curve in which the ordinates are the meter coefficients and the abscissas are the quantity discharged in cubic feet per second. The curve shows that the value of the discharge coefficient varies from 0.965 to 0.992, rising rapidly from the lowest quantity measurement of 0.08 cfs to 0.3 cfs, and then more gradually to the highest value at the maximum discharge of 1.6 cfs. These values were obtained by a graduate student in 1935 and checked by three other graduate students in 1936.

In Fig. 4 of this discussion is given a comparison between the writer's rounded-entrance meter and the conventional type. The dashed-line curve is for the 8.060 × 3.3759-in. bronze meter which the author used, and the solid line is for the writer's meter. The writer used the author's experimental results as given in Fig. 26 of the paper, because that particular test came closest to the conditions under which the writer's meter was rated. In order to put both instruments on the same basis the coefficients have been plotted, in Fig. 4 of this discussion, against $V \times D$ (velocity × diameter) which may be taken as the Reynolds number since the temperature of the water was held constant in both cases. In the meter having straight converging sides, the coefficient comes up to a maximum value and remains constant, but in the rounded-entrance venturi, the curve continues to rise.

Attention should be called to the fact that some textbooks express the formula for the quantity discharged by a venturi meter in terms of the area at the entrance. Since the meter is much more sensitive to changes in area at the smallest section, the formula should always be given as a function of the throat area.

Author's Closure

Professor Angus points out quite correctly that there should be a length of at least 10 or 12 diameters of straight pipe before the meter; nevertheless, many installations must and are made under most unfavorable conditions. Figs. 5 and 6 of this discussion give the test results obtained with the illustrated installa-

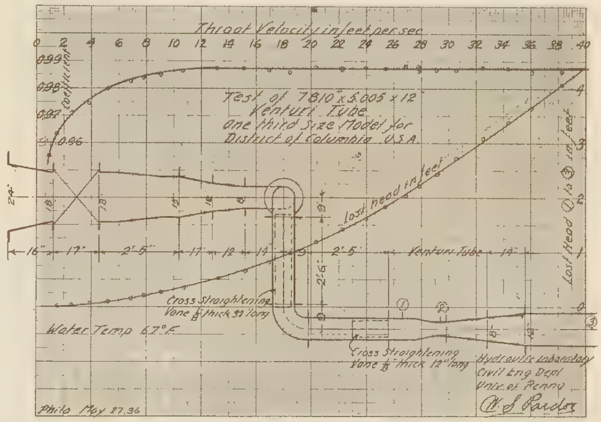

Fig. 5 Results of a Test of a 7.810 × 5.005 × 12-In. Venturi Meter

(This setup is a one-third size model of an installation at the District of Columbia.)

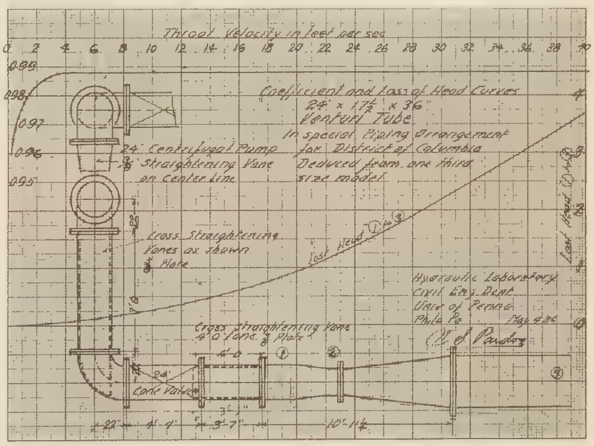

Fig. 6 Coefficient and Loss-of-Head Curves for a 24 × $17^1/_2$ × 36-In. Venturi Meter

(Special piping arrangement of an installation at the District of Columbia. Results deduced from one-third size model.)

tion which is a one-third size model of a large venturi meter at the District of Columbia. The straightening vanes shown effectively restored the normal coefficient in the model setup. The coefficient and the elliptical velocity distribution were used only to explain mathematically the difference between the normal coefficient and the coefficient for "shooting flow."

The author has always used C_r, as suggested by Professor Angus, when the inlet traverse is normal, as it usually is. Quite recently, the author used this method in computing a set of

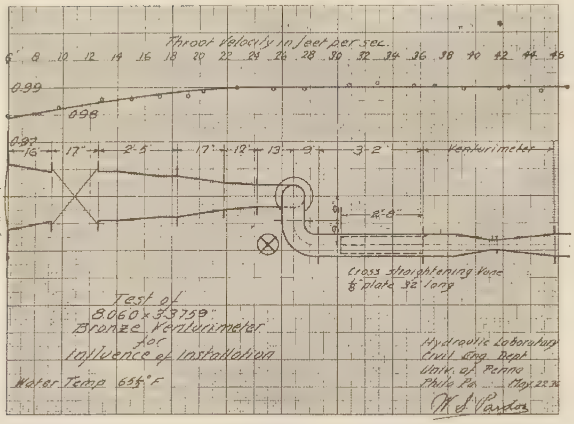

FIG. 7 TEST OF 8.060 × 3.3759-IN. BRONZE VENTURI METER FOR INFLUENCE OF INSTALLATION

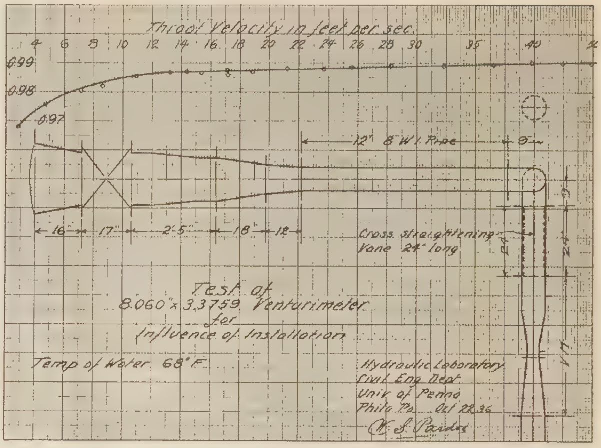

FIG. 10 TEST OF 8.060 × 3.3759-IN. VENTURI METER FOR INFLUENCE OF INSTALLATION

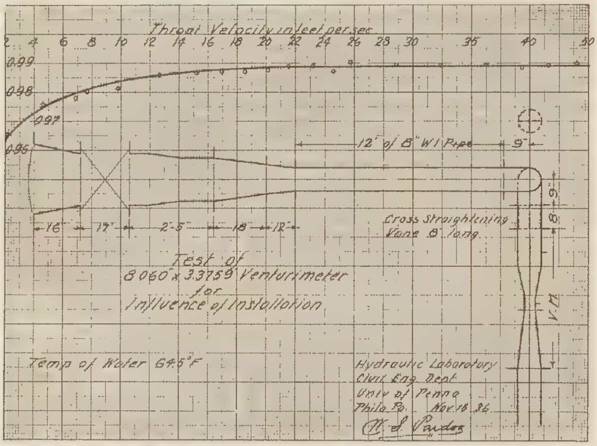

FIG. 8 TEST OF 8.060 × 3.3759-IN. VENTURI METER FOR INFLUENCE OF INSTALLATION

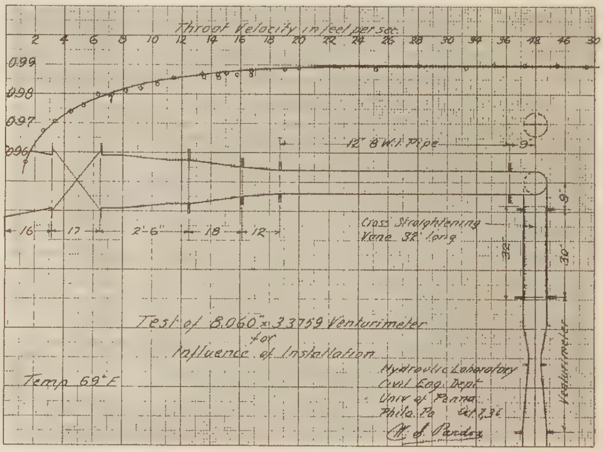

FIG. 11 TEST OF 8.060 × 3.3759-IN. VENTURI METER FOR INFLUENCE OF INSTALLATION

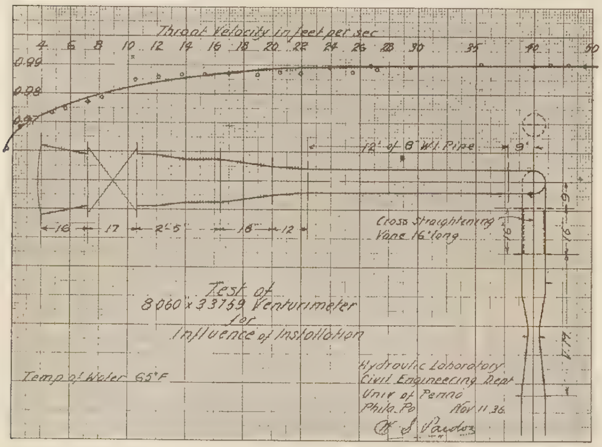

FIG. 9 TEST OF 8.060 × 3.3759-IN. VENTURI METER FOR INFLUENCE OF INSTALLATION

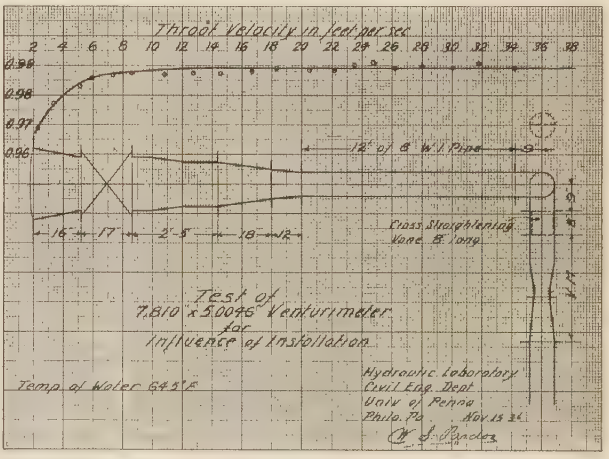

FIG. 12 TEST OF 7.810 × 5.0046-IN. VENTURI METER FOR INFLUENCE OF INSTALLATION

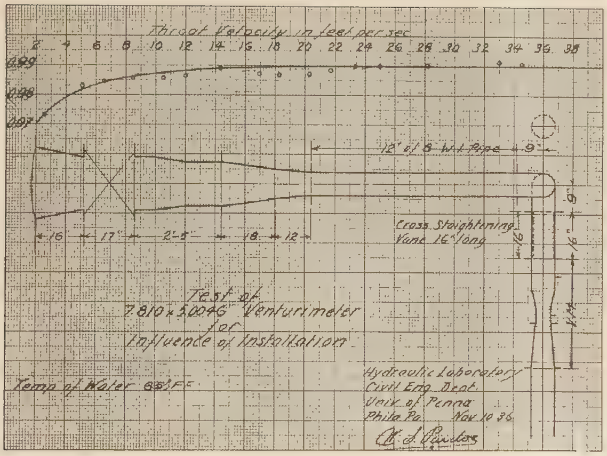

FIG. 13 TEST OF 7.810 × 5.0046-IN. VENTURI METER FOR INFLUENCE OF INSTALLATION

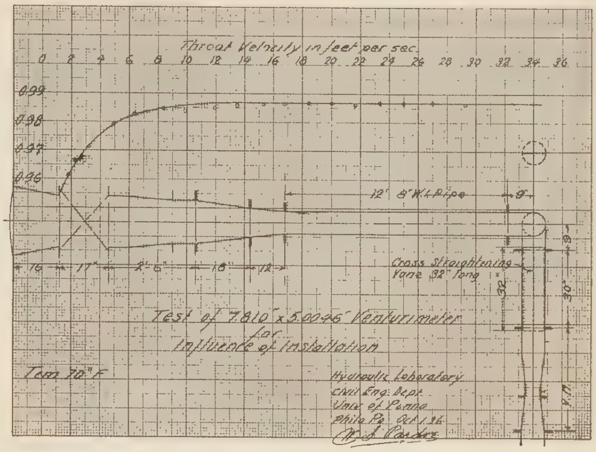

FIG. 15 TEST OF 7.810 × 5.0046-IN. VENTURI METER FOR INFLUENCE OF INSTALLATION

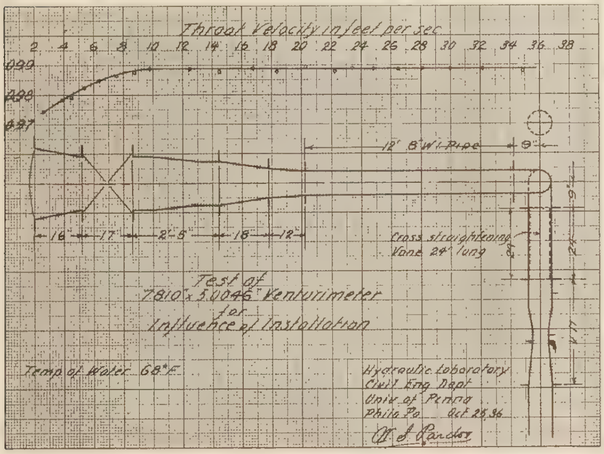

FIG. 14 TEST OF 7.810 × 5.0046-IN. VENTURI METER FOR INFLUENCE OF INSTALLATION

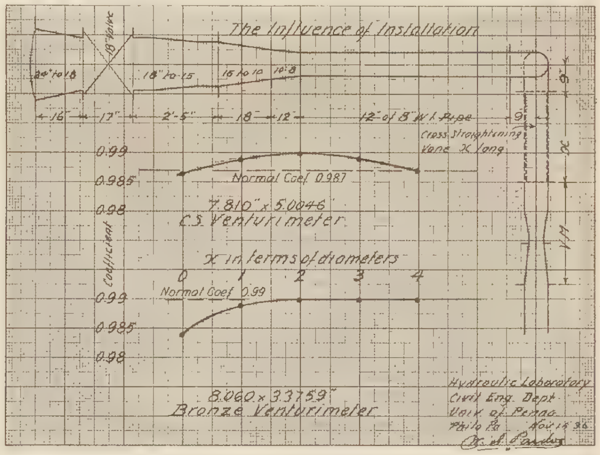

FIG. 16 SUMMARY OF RESULTS SHOWN IN FIGS. 7 TO 15, INCLUSIVE

coefficient curves for the A.S.M.E. Fluid Meters Report from coefficient curves of over 50 meters and finds that

$$C_r = 0.045/d_2^{0.23}$$

for $d_2/d_1 = 0.5$. In this equation d_2 is expressed in inches.

R. B. Smith's discussion of the effect of upstream roughness is most enlightening and his comparisons with the V.D.I. nozzle surprisingly confirmative. Anything which lowers the traverse coefficient (such as roughness, an elbow, or an enlarger) increases the coefficient of the venturi meter. Mr. Smith's mathematical treatment of vortex flow is very conclusive; it checks the experimental work much closer than the experimenter could reasonably expect. At some time in the future, the author will check the following statement which appears in the last paragraph of Mr. Smith's discussion: "The effect of helical flow, like other disturbances, is less severe with small ratios." An examination of Fig. 31 of the paper does not confirm this, but other factors enter and might control it.

Ed S. Smith is meticulous about the use of the term "shooting flow" and then uses the term "uniformly distributed flow." Uniform flow means something entirely different. What is meant is a traverse coefficient or pipe factor of unity at the main section; it would seem entirely reasonable that as the ratio d_2/d_1 approaches unity, the throat traverse coefficient would be less than unity.

The following answers are given the questions in the last paragraph of Ed S. Smith's discussion: (1) The value of g used was 32.16, which is correct for the latitude of Philadelphia, Pa. (2) The pressure at the throat was at all times above the atmospheric pressure and therefore far above any possibility of cavitation. (3) The low-ratio meter was a smooth brass casting with a bituminous paint from the upstream flange to a short distance above the throat which is smooth finished. The high-ratio meter was made of smooth cast iron with a coating of aluminum paint; it was of the usual construction with a finished brass throat.

Mr. Sprenkle mentions some tests by H. Mueller in which venturi meters were placed next to two 90-deg elbows in planes at right angles, such as shown in Fig. 31 of the paper, which resulted in a decrease in the coefficient of 0.25 per cent. This seems reasonable, but he also implies that when the meter was placed after a single 90-deg elbow it decreased the coefficient. Fig. 23 of the paper indicates the reverse effect.

Professor Sorenson's data on a 5 × 2$^1/_2$-in. venturi is of great interest since it gives a rising coefficient curve which of course suggests it would plot along with similar venturi meters in a satisfactory curve against Reynolds' numbers. If this curve,

shown in Fig. 4 of this discussion, were continued, it would appear to go above unity, which seems unreasonable. The combined plots do not vary more than 0.5 per cent, which is very good.

After preparing the paper, the author ran a series of experiments on the effectiveness of cross straightening vanes, the results of which are shown in Figs. 7 to 15, inclusive, and summarized in Fig. 16 of this discussion. It is evident from these figures that cross straightening vanes, four diameters long, will remove the effect of the vortex, but the effect of the low pipe coefficient is still present in the low-ratio meter up to three diameters. This is, of course, a great improvement over the ineffectiveness of straight pipe without straightening vanes, as shown in Fig. 31 of the paper.

Modified I.S.A. Orifice With Free Discharge[1]

J. E. Christiansen.[2] "End-cap" orifices possess certain definite advantages for making field tests on irrigation pumping plants especially for initial or acceptance tests, but heretofore their use has been limited by a lack of knowledge regarding correct coefficients to use. The first orifices of this type to be generally used for irrigation-pump testing were made and calibrated by manufacturers of turbine pumps. Some of these orifices were made from cast aluminum and were calibrated and stamped with a constant K to be applied in the formula

$$G = K A\sqrt{2gH}$$

in which G = gallons per minute
K = constant for a given orifice
A = area of orifice in square inches
H = head above center of orifice in inches

The head on the orifice was determined by drilling and tapping a hole ($^1/_8$ in. or $^1/_4$ in. pipe size) in the side of the pipe, at least 2 ft from the end, into which a short nipple was screwed. A short piece of rubber tubing was slipped over this nipple and a piece of glass tubing inserted in the other end. The glass tubing could then be held in such a position that the free water surface was visible so that the height above the center of the orifice could be measured.

Precise results were not expected from such measurements but it is doubtful if the magnitude of possible errors was recognized. Care was not always taken to remove the burr from the hole drilled for the piezometer connection, or to see that the nipple was set flush with the inside wall of the pipe. The orifice edges were sometimes rounded from pumping sand.

For concentric pipe orifices the writer has always used the formula

$$Q = \frac{C_1 A\sqrt{2gH}}{\sqrt{1-(d/D)^4}}$$

which is basically the same as for venturi tubes and nozzles. Available data on orifices indicated that the value of the coefficient C_1 remained practically constant for rather wide variations in the ratio d/D. The writer attempted to determine a probable value of C_1 from discharge curves for end-cap orifices published in a pump catalog, and from the constants K stamped on three orifices of different sizes.

The coefficients C_1 computed from the discharge curves for 12 orifices for casing varied from 0.597 to 0.664. Six of these orifices had ratios d/D varying from only 0.767 to 0.796; but the computed coefficients C_1 varied from 0.597 to 0.640. The coefficients computed from similar curves for five other orifices for standard pipe varied from 0.613 to 0.652, although the ratios d/D varied from only 0.770 to 0.783. The coefficients for three orifices for casing computed from values of K stamped on the orifices varied from 0.610 to 0.629, with the ratios d/D varying from only 0.782 to 0.796. Although there was no correlation between these coefficients and the ratio d/D, there appeared to be some relation between the coefficients and the diameter of the orifices. All orifices with diameters less than 5 in. had coefficients of 0.626 or more; and all orifices with diameters greater than 5 in. had

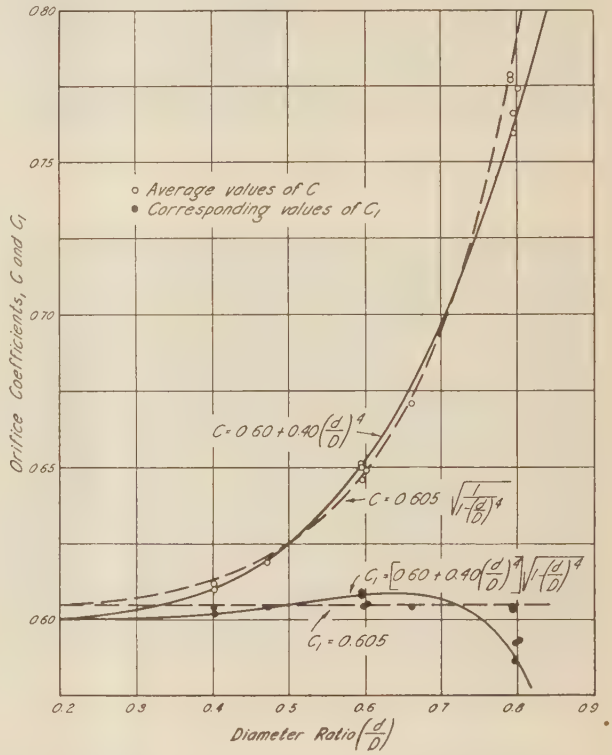

Fig. 1 Comparison of the Authors' Test Data With Different Formulas for the Coefficients C and C_1

coefficients of 0.629 or less. The highest coefficient 0.664 was for a $1^3/_4$-in. orifice with d/D = 0.1825. Obviously, reliable coefficients for orifices of various sizes and diameter ratios could not be obtained from this analysis.

The writer has computed the values of C_1 corresponding to values of C as given by the formula

$$C = 0.60 + 0.40\ (d/D)^4$$

and also values of C corresponding to C_1 = 0.605. The relationships are given by the expressions

$$C_1 = [0.60 + 0.40\ (d/D)^4]\sqrt{1-(d/D)^4}$$

$$C = \frac{0.605}{\sqrt{1-(d/D)^4}}$$

The comparison is shown graphically in the accompanying Fig. 1. Values of C_1 corresponding to the authors' average values of C for sharp-edged orifices and for pipe lengths of 19 diameters or more, as given in Table 1, were also computed and are shown

[1] Published as paper RP-51-1, by M. P. O'Brien and R. G. Folsom, in the January, 1937, issue of the A.S.M.E. Transactions.

[2] Assistant Irrigation Engineer, College of Agriculture, University of California, Davis, Calif.

in Fig. 1, which emphasizes the desirability of using diameter ratios less than 0.80. A constant value for C_1 would certainly not hold for higher ratios, although it agrees closely with the authors' expression for C for diameter ratios less than 0.75. It would appear from this comparison that a value of $C_1 = 0.605$ fits the experimental data as well as the proposed formula for C. The writer believes that engineers generally will find it more convenient to use a coefficient that is constant than one which varies with the diameter ratio.

Authors' Closure

The authors are indebted to Mr. Christiansen for his brief historical notes regarding the use and development of free-discharge orifices. It was the uncertainty as to coefficients which led to the investigations reported by the authors.[1]

The I.S.A. orifices and nozzles are constructed with corner taps to obviate the difficulties involved in drilling pressure taps, especially in the field. Both the orifice and holder, including the pressure slots, are constructed with shop accuracy and this arrangement is believed to increase the reproducibility and reliability of the results.

Choice of the discharge equation is basically a matter of convenience but there is an advantage in following previous work. The I.S.A. Standards mention only coefficients computed from the formula: $Q = CA\sqrt{(2gh)}$. The recent A.G.A.-A.S.M.E. report[3] bases the coefficient on the same formula for flange and pipe taps but uses the venturi formula for vena-contracta taps. As noted by Mr. Christiansen, the engineers who developed the "end-cap" orifice preferred the simpler form. The formula $0.60 + 0.40\,(d/D)^4$ was mentioned because of its simplicity but is not as precise as an average line drawn through experimental points, especially at large diameter ratios.

The authors failed to mention previously that the orifices and holders used in the tests reported in the paper were donated by the Byron Jackson Company.

The Collection and Evaluation of Data for the Design of Steam-Generating Units[1]

H. Kreisinger.[2] Mr. Cross calls attention in his paper to the radiation error when the temperature of hot gases in a boiler setting is measured with a thermocouple, and states that the smaller the wire of the couple used in the measurements the closer is the measured temperature to the true temperature.

This feature has been treated in a paper[3] which includes the results of numerous gas-temperature measurements with different sizes of thermocouples. These results show that small wire thermocouples give readings closer to the true temperature of gases than large couples, and that by the use of two or three couples of different wire sizes the true temperature can be approximated by extrapolation to zero-diameter wire. The physical reason for this behavior is that, under the usual conditions of temperature measurements, the thermocouple receives heat from gases by convection and loses heat to the surrounding colder surfaces by radiation. The surface that receives heat by convection is the surface of the gaseous film surrounding the hot junction of the thermocouple wire, whereas the heat is radiated directly from the metal surface of the thermocouple wire. The thickness of the gaseous film around the thermocouple wire remains nearly the same no matter what the size of the thermocouple wire, so that as the wire of the couple becomes smaller the surface of the couple giving up heat by radiation decreases faster than the surfaces of the gaseous film receiving heat by convection. As the diameter of the thermocouple wire approaches zero, the surface giving up heat by radiation also approaches zero but the surface of the gaseous film approaches a constant greater than zero. This relation is expressed by the equation

$$C\,(T - T_1)\,(D + 2_d) = (T_1^4 - T_2^4)\,D$$

Where

C is a constant for any given set of conditions.
T is temperature of gases
T_1 is temperature of the hot junction of the couple
T_2 is average temperature of the surfaces surrounding the stream of gases and receiving heat from the thermocouple by radiation
D is diameter of the hot junction wire of the couple
d is thickness of gaseous film around the thermocouple wire.

Fig. 1

The left side of the equation is proportional to the heat imparted to the hot junction by convection, and the right side is proportional to the heat lost by radiation. As D approaches zero, $(T - T_1)$ also approaches zero; that is, T_1 becomes equal to T.

The method of using two or three thermocouples of different wire sizes for arriving at the true temperature of gases is applicable only for clean gases, or dusty gases at sufficiently low temperature so that the dust particles do not stick to the exposed hot junction of the thermocouple. With pulverized coal and probably with stoker firing there is so much ash and coke particle carried by the gases that the exposed hot junction of the couple becomes coated with slag in about three minutes when the temperature of gases is higher than 1800 F. The temperature indicated by the couple drops rapidly at first and becomes nearly constant at the end of three minutes' exposure. The following readings taken at a point where the gases enter the boiler show the temperature drop during the first three minutes of exposure.

Time after insertion of couple	Temperature, deg F
0	2370, 2355, 2325
1/2	2250
1	2235
1 1/2	2175
2	2175
2 1/2	2160
3	2145

At the end of three minutes the slag accumulation on the couple forms a ribbon about 1/8 in. wide in front of the thermocouple wire as shown in Fig. 1. The drop in temperature with the accumulation of slag may be due to the following causes:

(*a*) Slag keeps hot gases away from the wire, thereby reducing the heat imparted by the gases by convection

(*b*) Slag screens the wire from receiving heat by radiation from hotter parts of the furnace and leaves the wire surface radiating heat to cooler boiler surface unchanged

(*c*) Particles of coke which are deposited with ash on the couple burn, and the heat so generated raises the temperature of the couple immediately after the insertion of the couple. As the

[3] "History of Orifice Meters and the Calibration, Construction, and Operation of Orifices for Metering," Joint A.G.A.-A.S.M.E. Report, The American Society of Mechanical Engineers, 29 West 39th Street, New York, N. Y., 1936.

[1] Published as paper PRO-59-4, by B. J. Cross, in the February, 1937, issue of the A.S.M.E. Transactions.

[2] Engineer in charge research and development, Combustion Engineering Co., New York, N. Y. Mem. A.S.M.E.

[3] "Radiation Error in Measuring Temperature of Gases," by H. Kreisinger and J. F. Barclay, Trans. A.S.M.E., vol. 39, 1917, p. 107.

slag ribbon on the couple becomes wider the particles of burning coke are farther away from the wire and are insulated from it by the slag already deposited, and the heat generated is dissipated in the gases. It takes only about 3 seconds for these coke particles to burn completely after they are deposited.

The high temperatures immediately after the insertion of the couple may be used as indication of the quantity of coke carried by the gases. If the quantity of coke is small the heat generated by the burning of the coke may compensate the heat lost by radiation and the reading may be very close to the true temperature of the gases. If, however, the quantity of coke is large the temperature indicated by the couple is higher than that of the gases. This latter condition may prevail if the temperature measurement is made in the furnace nearer the burners where the combustion is far from completion. Ordinarily the coal particles stay in the furnace only one to two seconds. Those particles that stick to the thermocouple stay in the furnace long enough to be completely burned.

If the explanation under (*c*) is correct the results of temperature measurements in the furnace may be several hundred degrees higher than the true average temperature of gases. For this reason any temperature measurements in the furnace should be made as far from the burners as possible where the combustion is nearly completed in order that the error may be small.

Temperatures after the superheater are usually too low for ash particles to stick to the couple, and the wire remains clean. The temperature measurements when taken with small couple are near the true temperature of gases.

If the hot junction of the couple is made of wire 0.007 in. in diameter and the indicated temperature is 1200 F the true temperature is not over 1250 F and may be 1225 F.

With clean gaseous fuels such as natural gas, the products of combustion are free from dust particles and the measurement of temperature is a comparatively simple problem.

The preceding discussion shows that the measurements of temperature in boiler furnaces and settings require great care, and much judgment is needed in the interpretation of the results. The designer of the superheater must have fairly accurate data on the temperature of the products of combustion entering the superheater, and the rate of heat transfer if he is to design a superheater giving the desired superheat within the limits of tolerance.

Author's Closure

As Mr. Kreisinger points out, the greatest difficulty in the measurement of furnace temperature is due to the accumulation of slag on the hot junction of the couple. Such accumulation may cause low readings because of the insulating effect of the slag, or if there were unburned carbon in the furnace dust the first effect of the slagging may possibly be to cause a high reading for a short period due to the combustion of the carbon on the wire.

Mr. Kreisinger presents a logical explanation on the effect of wire diameter in the measurement of gas temperatures in the presence of cooler surrounding surfaces. There can be no question but that the smaller the wire of the hot junction, the closer will be the measured temperature to the true temperature. For field measurements, when a large number of point temperatures must be measured, the wire should be of such size that it is not too fragile. It should also be possible to make up new junctions quickly by welding and to attach them by welding to the couple lead wire. These operations must be done in the field in a minimum of time and without dismounting the couple. Wire of 0.007 in. diameter has been found suitable to these requirements.

To one interested only in scientific accuracy, the measurement of temperature of furnace gases might appear to present insurmountable difficulties. For purposes of control in furnace operation and for design of boilers and superheaters, absolute accuracy in the measurement of temperature is not essential. The operator of boiler furnaces needs only a reference temperature, above which he may get into trouble from slag accumulation or slag erosion. The designer of heat-absorbing equipment needs a reference temperature to which he can relate the performance of his equipment. Such reference temperature should always be measured by the same method and in the same manner. While, in the measurement of this temperature, efforts should be made to approach the true temperature as closely as possible, a consistent variation from the true temperature would not be serious.

Performance of Lubricants Based on Diesel-Engine Service Conditions[1]

I. I. Sylvester.[2] The writer's comments do not refer particularly to the paper, but rather are an effort to develop some thought regarding the features of an analysis of a lubricating oil which should govern the length of its service. In this respect there does not appear to be any standard unit of measurement, either in hours or miles, which can be taken as a limit for the use of a lubricating oil. It is quite apparent that such a limit will vary greatly depending on the type of service, i.e., road or switching service, wherein the limit is quoted in miles and hours, respectively. The Canadian National Railways obtain an average of approximately 14,000 miles per change of lubricating oil in rail cars from 200 to 400 hp.

There appears to be a real need for determining the items in a lubricating-oil analysis which change with use, and for developing standards which will indicate how much change will be tolerated in these various items before it is necessary to remove the oil from the engine. Discussion or work along this line will certainly be of use to those deciding when the oil should be removed.

In Table 2 of the paper, giving data on cylinder-liner wear, the measurements were taken from the top of the piston. It is the practice on the Canadian National Railways, when measuring cylinder-liner wear, to record the wear at the point of the top-ring travel. In making comparisons as in the author's Table 2, it is important to record the wear of the cylinder liner at the top-ring travel point. Other measurements often mislead.

T. B. Rendel.[3] The writer agrees with the author regarding the effect of faulty or slow combustion of the fuel on the sludge formation of lubricating oil. When faced with a complaint of sludge formation or ring sticking due to supposedly faulty lubricating oil, it is the writer's practice to examine the fuel oil at once to determine if it is at fault and causing poor combustion. In this connection, the writer has often wondered whether or not the type of lubricating oil which will prevent piston-ring sticking in a Diesel engine is just the opposite type of lubricating oil which will accomplish the same purpose in a gasoline engine.

In the latter case we need an oil which resists sludge formation to the best possible degree. Such an oil, however, usually has no solvent power for sludge-like materials; and this solvent power is just what is needed in a Diesel engine to remove sludge formed by poor combustion of fuel and to prevent it from coking on and behind piston rings, causing them to stick. While true that such an oil would result in fairly large quantities of sludge in the main body of the crankcase, it could be dealt with there by efficient filters or, in extreme cases, by centrifuges.

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[1] Published as paper OGP-59-3, by C. M. Larson, in the February, 1937, issue of the A.S.M.E. Transactions.

[2] Special Engineer, Operating Department, Motive Power and Car Equipment Sections, Canadian National Railways, Montreal, Canada.

[3] Shell Petroleum Corporation, Shell Building, St. Louis, Mo.

Indexes to A.S.M.E. Papers and Publications

THE following pages will serve as a guide to the current publications of the A.S.M.E. during the calendar year 1937, and also to publications developed by the technical committees. The publications of the Society may be classified in two general groups, regular and special, as follows:

REGULAR SOCIETY PUBLICATIONS, 1937

Mechanical Engineering, monthly (see index on pages RI-101–113)
A.S.M.E. Transactions, monthly (see index on pages RI-115–127)
Mechanical Catalog, 1937–1938 edition.

SPECIAL PUBLICATIONS ISSUED IN 1937

1936 Oil Engine Power Cost Report
Hydraulic Structures, published September, 1937
1936 Proceedings of Graphic Arts Technical Conference, published May, 1937
Biography of James Hartness by Joseph W. Roe, published July, 1937

Standards

Machine Tapers, approved March, 1937, published March, 1937
Adjustable Adapters, approved December, 1937, published December, 1937
Large Rivets ($^1/_2$ inch diameter and larger), approved March, 1937, published March, 1937
Addendum to Cast-Iron Long Turn Sprinkler Fittings, approved December, 1937, published December, 1937

Power Test Codes

Instruments and Apparatus, Part 15, Measurement of Surface Areas, approved July, 1935, published March, 1937

Safety

Safety Code for Elevators (revision) approved July, 1937, published September, 1937
Elevator Inspectors' Manual, approved July, 1937, published September, 1937

Boiler Code

A.S.M.E. Unfired Pressure Vessel Code, published September, 1937
Power Boiler Code, including Material Specifications, published, December, 1937

Research

Report of the Joint AGA-ASME Committee on Orifice Coefficients (reprint of 1935 report)
Fluid Meters—Part 1, Their Theory and Application (Fourth Edition) published August, 1937
Supplement to Bibliography on Management Literature (1931–1935), published July, 1937

Papers Presented at A.S.M.E. Meetings, 1937

The complete technical programs of the meetings of the Society and of its Professional Divisions have been published in *Mechanical Engineering* and may be located by consulting the index on pages RI-101–113. A considerable number of papers and reports included in these programs were not published during the year in either Transactions or *Mechanical Engineering*, but were issued in mimeographed or photo-offset form. Complete sets of these are on file for reference purposes at the office of the Society and the Engineering Societies Library, under the title of "Miscellaneous Papers Presented at A.S.M.E. Meetings, 1937." Photostatic copies of any of the papers may be secured from the Library at regular rates. A list of these papers and reports follows.

Miscellaneous Papers Presented at A.S.M.E. Meetings, 1937

Baker, Howard, Impression Lead and Electrotype Foil—Their Manufacture and Use by the Electrotyper
Boas, R. H., Rehabilitation of Station No. 3 Rochester Gas & Electric Corp., Rochester, N. Y.
Boller, E. R., Nitric-Acid Etching of Photoengravers' Zinc
Bonnet, F., The Manufacture and Uses of Cut Rayon Staple
Boyer, Glenn C., Waste-Heat Recovery From Diesel Engines
Bristol, E. S., and Peters, J. C., Some Fundamental Considerations in the Application of Automatic Control to Continuous Processes
Carpenter, Lewis V., Mechanical Engineering in Sanitation
Chandler, J. S., Penn State Method of Diesel-Fuel Testing
Duer, J. V. B., Place of Railroads in Modern Transportation
Duhring, E. L., Quick-Drying Inks
Dunn, J. C., Progress and Problems in Printing Rollers
Epstein, Samuel, Constitution and Properties of Type Metals
Exline, Paul G., Pressure-Responsive Elements
Fehse, Rudolph, Photogelatin Color Printing
Finkeldey, W. H., Sheet Zinc for Photoengraving
Freedman, Ephraim, The History of Rayon in the Retail Field
Hinkle, G. W., Stainless Steels and Their Application in the Textile and Rayon Industries
Holser, E. F., Industrial Heating and Process Furnaces
Hubner, W. H., and Egloff, Gustav, A Study of Diesel Fuels
Isaacson, A. M., An Investigation of Wood Sanding with Abrasive Belts
Kantrowitz, M. S., and Simmons, R. H., The Technical Status of Permanence and Durability of Paper
Ketcham, Howard, Color Comes of Age
Lewis, L. L., Air Conditioning for Textile Plants Making and Using Synthetic Yarns
Livingston, L. F., Processing Engineering in Agriculture
Lund, N. B., Industrial Experience as Applied to Sanitation
MacArthur, Charles, Rapid-Drying Inks on Web-Fed Papers
McGar, B. H., Sheet Brass for Photoengraving and Etching
Meyer, F. H., Economical Application of Lumber Lifts
Passano, W. M., Employer-Employee Relations—Wages
Report on Oil-Engine Power Cost for 1936
Schwarz, Ralph H., and Winkler, J. Homer, Heat-Treated Electrotypes
Smith, Lybrand P., Cavitation on Marine Propellers (Additional data for) paper published in July, 1937, Transactions of the A.S.M.E.)
Spencer, Ernest W., The Technical Qualifications of Photoengraving Zinc—Statements by Manufacturers and Operators
Taylor, Henry W., The Mechanical Background of Refuse Incineration
Tegge, Albert R., Jr., A Method for Rating the Comparative Machinability of Woods
Van Valkenburg, Martin, Lumber and Veneer Waste—An Important Element of Cost in the Furniture Industry
Waters, E. O., Wesstrom, D. B., Rossheim, D. B., and Williams, F. S. G., Formulas for Stresses in Bolted Flanged Connections—Appendix "W A"
Weaver, S. H., The Creep Curves and Stability of Steels at Constant Stress and Temperature. Test Curves for Tables 1–4, published in Trans. A.S.M.E., Nov., 1936
Wellington, C. O., Where Are the Profits?

Publications Developed by the Technical Committees

THE Society's technical committees, the first of which was organized many years ago and all of which have been continuously at work on codes, standards, research, and other special reports, have developed a series of publications of permanent value to the membership. The following list is first presented here for record and for ready reference. This list covers the entire group of publications of these committees completed to date which are now available.

To assist the members in securing copies of these publications the sale price is also given. It should be recalled, however, that a discount of 20 per cent is allowed to members of the A.S.M.E. on all pamphlet publications, except standards and the cases specially noted.

RESEARCH

Dynamic Loads on Gear Teeth (1932), $1.50
Fluid Meters
- Part 1—Theory and Application (1937), $3.00
- Part 2—Description of Meters (1931), $1.75
- Part 3—Selection and Installation (1933), $1.50

Report of the AGA-ASME Committee on Orifice Coefficients (1935), $2.75 (no discount)
Tests on Electrical Equipment for Drilling Rotary Drilled Oil Wells (1933), $0.85
Tests on Steam Equipment for Drilling Rotary Drilled Oil Wells (1932), $0.85
Roll-Neck Bearings (1935), $1.50
Bibliography on Cuttings of Metals (1866–1930), $1.25
Bibliography on Deterioration of Condensing Equipment (1845–1930), $1.25
Bibliography on Effect of Temperature Upon Properties of Metals (1828–1931), $1.25
Bibliography on Management Literature and Supplement (1903–1935), $2.75
Bibliography on Mechanical Springs (1678–1927), $1.25
Bibliography on Woods of the World (1928), $1.25
Bibliography on Marketing Research (1935), $1.00

POWER TEST CODES

Test Codes for

Atmospheric Water-Cooling Equipment (1930), $0.45
Centrifugal and Rotary Pumps (1928), $0.50
Compressors and Exhausters (1935), $0.95
Condensing Apparatus (revision in preparation)
Displacement Compressors and Blowers (revision in preparation)
Evaporating Apparatus (1927), $0.50
Feedwater Heaters (1927), $0.35
Gas Producers (1928), $0.55
Internal-Combustion Engines (1930), $0.55
Liquid Fuels (1930), $0.35
Reciprocating Steam Engines (1935), $0.65
Reciprocating Steam-Driven Displacement Pumps (1927), $0.65
Refrigerating Systems (1927), $0.55
Solid Fuels (1931), $0.55
Speed-Responsive Governors (1927), $0.45
Stationary Steam-Generating Units (1936), $0.60
Steam Locomotives (1927), $0.55
Steam Turbines (1928), $0.55
Hydraulic Prime Movers (revision in preparation)

Supplementary Codes and Publications

Definitions and Values (1931), $0.40
General Instructions (1929), $0.35
Instruments and Apparatus—
- Part 1—General Consideration (1935), $0.35
- Part 2—Pressure Measurement; Chapter 1, Barometers, and Chapter 6, Tables, Multipliers and Standards (1929), $0.35
- Part 2—Pressure Measurement; Chapter 2, Static and Total Pressure, Static Holes and Tubes, Impact Tubes, and Chapter 3, Pipes for Pressure Measurement (1936), $0.65
- Part 3—Temperature Measurement; Chapter 1, General; Chapter 5, Pyrometric Cones; Chapter 6, Liquid-in-Glass Thermometers; and Chapter 7, Bourdon Tube Thermometers (1931), $0.75
- Part 3—Temperature Measurement; Chapter 2, Radiation Pyrometers (1936), $0.55
- Part 3—Temperature Measurement; Chapter 8, Optical Pyrometers (1933), $0.35
- Part 4—Head Measuring Apparatus (1933), $0.35
- Part 6—Electrical Measurements (1934), $1.25
- Part 9—Heat of Combustion (1932), $0.40
- Part 10—Flue and Exhaust Gas Analyses (1936), $1.35
- Part 11—Determination of Quality of Steam (1931), $0.45
- Part 12—Measurement of Time (1932), $0.35
- Part 13—Speed Measurements (1930), $0.45
- Part 14—Linear Measurements (1936), $0.55
- Part 15—Measurement of Surface Areas (1937), $0.75
- Part 16—Density Determinations (1931), $0.30
- Part 17—Determination of the Viscosity of Liquids (1931), $0.45
- Part 18—Humidity Determinations (1932), $0.50
- Part 20—Smoke-Density Determinations (1936), $0.65
- Part 21—Leakage Measurement, Chapter 1, Condenser Leakage Tests (1928), $0.35
- Part 21—Leakage Measurement; Chapter 2, Boiler and Piping; Chapter 3, Steam Engine Leakage (1932), $0.35

BOILER CODE

Power Boiler Code (1937) Including Materials Specifications, $2.50
Locomotive Boiler Code (1935) with 1936 and 1937 Addenda, $0.55
Low-Pressure Heating Boiler Code (1935) with 1936 and 1937 Addenda, $0.65
Miniature Boiler Code (1935) with 1936 and 1937 Addenda, $0.50
Suggested Rules for Care of Power Boilers (1935), $0.70
A.S.M.E. Unfired Pressure Vessel Code (1937), $0.75
A.S.M.E. Boiler Construction Code, Combined Edition (1937 Edition in preparation)
Boiler Code Interpretation Sheets, $1.25 per set with binder. Single sheets, $0.15
- Annual Subscription, $2.50

STANDARDS

Machine Shop Practice Standards

Shafting and Stock Keys (B17.1—1934), $0.45
Code for Design of Transmission Shafting (B17c—1927), $0.75
Woodruff Keys, Keyslots, and Cutters (B17f—1930), $0.35
Tolerances, Allowances, and Gages for Metal Fits (B4a—1925), $0.50
American Standard Screw Threads for Bolts, Nuts, Machine Screws, and Threaded Parts (B1.1—1935), $0.60
Wrench-Head Bolts and Nuts and Wrench Openings (B18.2—1933), $0.50
Slotted Head Proportions; Machine Screws, Cap Screws, and Wood Screws (B18c—1930), $0.45
Track Bolts and Nuts (B18d—1930), $0.40
Round Unslotted Head Bolts, Carriage, Step, and Machine Bolts (B18e—1928), $0.40
Plow Bolts (B18f—1928), $0.35
Tinners', Coopers', and Belt Rivets (B18g—1928), $0.35
Rotating Air Cylinders and Adapters (B5.5—1932), $0.35
Jig Bushings (B5.6—1935), $0.35
T-Slots, Their Bolts, Nuts, Tongues, and Cutters (B5a—1927), $0.35
Tool Holder Shanks and Tool Post Openings (B5b—1929), $0.25
Milling Cutters (B5c—1930), $0.75
Taps—Cut and Ground Threads (B5e—1930), $0.50
Spur Gear Tooth Form (B6.1—1932), $0.45
Gear Materials and Blanks (B6.2—1933), $0.50
Shaft Couplings (B49—1932), $0.35
Drawings and Drafting Room Practice (Z14.1—1935), $0.50
Small Rivets (B18a—1927), $0.30
Socket Set Screws and Socket Head Cap Screws (B18.3—1936), $0.40
Circular and Dovetail Forming Tool Blanks (B5.7—1936), $0.40
Chucks and Chuck Jaws (B5.8—1936), $0.45
Lathe Spindle Noses (B5.9—1936), $0.50
Large Rivets (B18.4—1937), $0.65
Machine Tapers (B5.10—1937), $0.50
Adjustable Adapters (B5.11—1937), $0.50

Pipe and Fittings

Wrought-Iron and Wrought-Steel Pipe (B36.10—1935), $0.50

Code for Pressure Piping (no discount) (B31.1—1935), $1.00
Cast-Iron Pipe Flanges and Flanged Fittings for 25 Lb Maximum Saturated Steam Pressure (B16b2—1931), $0.40
Cast-Iron Pipe Flanges and Flanged Fittings for 125 Lb Maximum Saturated Steam Pressure (revision in preparation)
Cast-Iron Pipe Flanges and Flanged Fittings for 250 Lb Maximum Saturated Steam Pressure (B16b—1928), $0.50
Cast-Iron Pipe Flanges and Flanged Fittings for 800 Lb Maximum Hydraulic Pressure (B16b1—1931), $0.35
Malleable-Iron Screwed Fittings for 150 Lb Maximum Saturated Steam Pressure (B16c—1927), $0.40
Cast-Iron Screwed Fittings for 125 and 250 Lb Maximum Saturated Steam Pressure (B16d—1927), $0.35
Steel Flanged Fittings and Companion Flanges for 150, 300, 400, 600, 900, 1500 Lb Maximum Steam Service Pressure (B16e—1932), $0.65
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SAFETY

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Safety Code for Mechanical Power-Transmission Apparatus (no discount) (B15—1927), $0.35

JOINT CODE

API-ASME Code for Unfired Pressure Vessels for Petroleum Liquids and Gases (third edition in preparation), (no discount) $1.00

Biographies

BIOGRAPHIES issued under the sponsorship of the A.S.M.E. Biography Advisory Committee are as follows:

Life of Henry Laurence Gantt, by L. P. Alford (1934), $5.00
Biography of John Stevens, by Archibald Douglas Turnbull (1928) $5.00
Autobiography of John A. Brashear (1924), $5.00
Biography of James Hartness, by Joseph W. Roe (1937), $4.00
Biography of Robert Henry Thurston, by William F. Durand (1929), $5.00
Biography of John Edson Sweet, by Dean A. W. Smith (1925), $4.50

Books on Special Subjects

Hydraulic Laboratory Practice (1929), $10.00
Bulletin 3, Munich Hydraulic Institute (1935), $3.75
Hydraulic Structures (1937), $18.00
1936 Oil Engine Power Cost Report (1937), $1.00

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